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# Heating Piping & Air Conditioning

INDUSTRIAL
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DESIGN
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MAY ★ 1947

Volume 103 — Journal Section of the American Society of Heating and Ventilating Engineers

O. W. Ott's article on page 75, "Getting Heat Down to the Floor", is but one of many practical discussions presented this month

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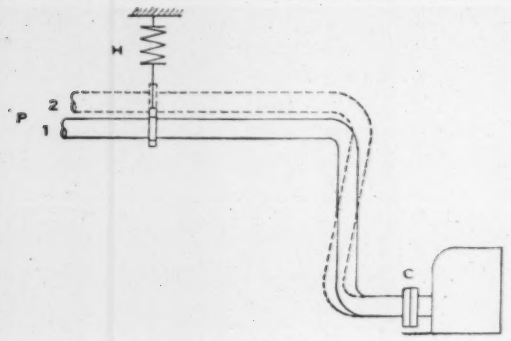
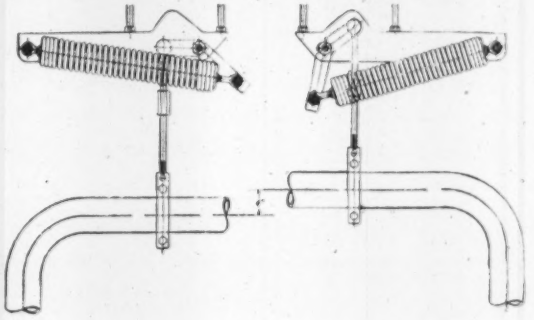
# Are Weight Transfer Stresses Endangering Your Piping System?

## THE PROBLEM:

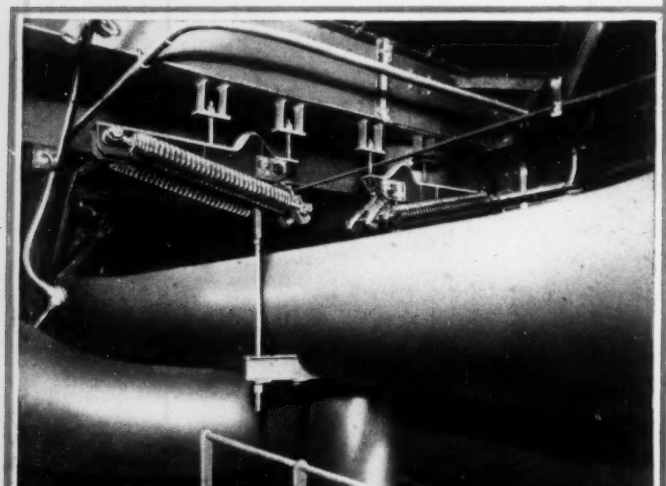



If the full safety factor is to be maintained in piping, then any weight transfer stresses caused by vertical movement of piping due to expansion or contraction, load change, surge or vibration must be eliminated. Diagram illustrates this problem—if pipe hangers "H" do not support full weight of the piping "P" in both cold position "1" and hot position "2", then dangerous stresses may result at connections "C".

## THE SOLUTION:



Provide constant support to the piping throughout the entire range of vertical movement with Grinnell Constant-Support Hangers—the *only* constant support hangers. A lever, turning on the main pivot, balances the weight of a vertically shifting load because the change in moment arm with lever rotation is such that the product of the moment arm and the spring force is always constant and equal to the weight of the piping system.



*Main steam line at Hell Gate Station of Consolidated Edison Company is provided constant support by Grinnell Constant-Support Hangers.*

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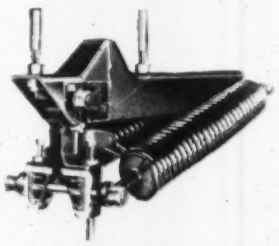
— the ONLY Constant-Support Hangers

**FEATURES:**

- Constant support of piping in all "Hot" and "Cold" positions.
- Full safety factor of the supported system is always maintained.
- Non-resonant, energy absorbing.
- Mass-produced from standard precision parts.
- Individually calibrated for each installation.
- Load-adjustment features incorporated into the design.



- Three models meet entire range of load-travel specifications.
- Minimum headroom required.

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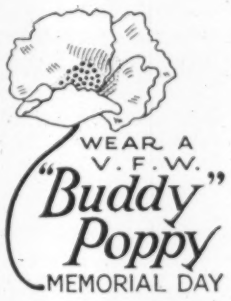
WHENEVER **PIPING** IS INVOLVE

# The Editor's Pages

## HAS CONSTRUCTION COST PEAK BEEN REACHED?

Fifty-three per cent of 268 general contractors in all parts of the country polled by F. W. Dodge Corp. believe that the peak in construction costs has been reached, it was reported last month.




In some areas, notably in New England, the South Atlantic states, and the East South Central states, opinion was preponderant, at least two to one, that costs had attained their highest level. Only in the Pacific states was opinion general that the peak had not been reached, and here further cost climbs ranging from 11 to 12 per cent during the current calendar year were expected.

Forty-seven per cent of the contractors believe that further increases can be expected, the average increase anticipated by them being 8.7 to 10.2 per cent above current levels.

The general contractors reported, by extremely high majorities, that the following unusual factors contribute to the high costs of building at present:

Labor inefficiency caused by irregular flow of materials.

Delays in construction owing to inadequate supply of skilled workers.

Excessive expenditure of time in shopping for and obtaining materials.

Many general contractors reported that the greatest contributing factors to high present-day costs were the low productivity of building mechanics, the low level of skill of some craftsmen, and the failure of workers to do "a day's work for a day's pay."

Respondents in general said that they expected improvement in these retarding factors, many in the next year or two, but the consensus was that improvement would come only when the supply of materials and the numerical and qualitative strength of the labor force was adequate to meet the demands for construction.

In the aggregate, the general contractors reported that 34.6 per cent of their present work, by dollar volume, is being done on a lump-sum basis, while the remainder is being done on cost-plus-fixed-fee contracts.

## REPORTS ON STATUS OF PIPING STANDARDIZATION

News about projects of the American Standards Association was given in Industrial Standardization, the ASA's journal, for March. Following are reports from this source on several piping standards:

Cast Iron Flanges and Flanged Fittings for Refrigerant Piping, 300 Lb, B16.16—This proposed standard, recently submitted to the American Standards Association for approval, was initiated originally in 1924 by a subgroup on ammonia flanges and fittings of subcommittee 1 of the sectional committee on pipe flanges and fittings. Data collected by the Refrigerating Machinery Association covering dimensions and sizes of the interchangeable features of ammonia fittings and valves were used in preparing a recommended practice which was adopted by the Refrigerating Manufacturers' Association in 1931, but was not referred to the sectional committee for action. Continued requests for the standardization of refrigerant flanges and fittings resulted in appointment of a new subgroup of subcommittee 1 for a restudy of the situation in 1943. The final report of the subgroup was completed in January 1945 and distributed broadly for criticism and comment. It has now been approved by the sponsors and submitted to ASA. The proposed new American Standard is being referred to the mechanical standards committee for approval.

Code for Pressure Piping, B31-1-1942—A proposed second supplement consisting of corrections, changes, and additions to the 1942 edition of the American Standard Code for Pressure Piping has been submitted to the American Standards Association for approval as American Standard. The first supplement was approved by the ASA in November 1944. Both supplements originated with the subcommittee on scope and intent which was appointed to handle inquiries concerning the various requirements of the code. The new supplement has been sent out to letter ballot of the mechanical standards committee.

Soldered Joint Fittings, A40.3-1941—Studies are now under way for a revision of the 1941 edition of this American Standard. A first draft is being circulated for criticism.

Stainless Steel Pipe and Fittings—The Heating, Piping and Air Conditioning Contractors National Association has requested the ASA to take the necessary steps to establish an American Standard for stainless steel pipe and fittings. This request was made following a conference between the committee on standards of the HPACCNA and principal manufacturers of stainless steel pipe and fittings.

The ASA mechanical standards committee has decided that the work on stainless steel fittings should be referred to the sectional committee on pipe flanges and fittings, B16, and the work on stainless steel pipe to the sectional committee on dimensions and materials of wrought iron and wrought steel pipe and tubing, B36.

## 72 SIGN FOR REFRIGERATION, AIR CONDITIONING SHOW

Seventy-two companies already have taken 112 exhibit spaces in the Cleveland public auditorium for the fifth "All-Industry" Refrigeration and Air Conditioning Exposition to be held January 26 to 29, according to an announcement last month by show headquarters, 1107 Clark Bldg., Pittsburgh.

These companies are members of the Refrigeration Equipment Manufacturers Association, sponsors of the show. On May 1 the exposition was opened to other manufacturers who had displays in the fourth exposition in Cleveland in October 1946. Later the show will be opened to all companies in the industry.

# The Editor's Pages

### "IT AIN'T NECESSARILY SO"

"Many of the data dealing with dust and soot fall that have been presented at public hearings have been invested with an accuracy and reliability which, when analyzed, is quite misleading. To associate them statistically with the number of deaths from pneumonia or tuberculosis and report the relation to the extent of correlation coefficients is unwarranted. The use of correlation coefficients should be practiced with judgment. One can get an excellent correlation between the growth of the office boy and the growth of the company's business." —*from a paper on Factors Rarely Considered in Smoke Abatement, by Henry F. Hebley, presented at the annual meeting of the ASME, December* 1946, *and published in Mechanical Engineering, April* 1947.

### CONVECTOR PRODUCTION RISES SHARPLY

Production achievements of the manufacturers of convectors, standardization of sizes of convectors, and the new testing and rating code for convectors were among the subjects discussed at the annual meeting of the Convector Manufacturers Association at Cleveland, held in connection with the exposition and convention of the American Society of Heating and Ventilating Engineers.

L. D. Mandell, C. A. Dunham Co., was elected president of the association. E. A. Cline, the Trane Co., was elected secretary-treasurer.

"The current rate of production of the convector industry is more than twice our pre-war rate," said W. R. McMahon, Modine Mfg. Co., in his report as retiring CMA president. "During the year 1946 there were 17.3 million sq ft of convectors produced, according to a report of the Civilian Production Administration. Production of convectors during the fourth quarter of 1946 amounted to 6.9 million sq ft. The convector industry expects still further gains in production in 1947."

Significant new developments in convectors such as the baseboard convector and the radiant convector were discussed at the meeting.

The new testing and rating code for convectors drawn up under the joint sponsorship of the Convector Manufacturers Association and the Institute of Boiler and Radiator Manufacturers with the assistance of the National Bureau of Standards was reviewed. Considerable progress is being made by manufacturers in the standardization of convector sizes. Further activity in this direction is contemplated.

### STANDARD GIVES ELECTRICAL SYMBOLS

A "master" standard covering symbols used in drawings employed in wide sectors of the electrical field is ready for distribution, according to the American Standards Association, 70 E. 45th St., New York 17. Known as Basic Graphical Symbols for Electric Apparatus, Z32.12-1947, it costs 40c.

The standard provides 152 basic symbols for electrical drawings, which in combinations can be used to describe on electrical diagrams an almost endless variety of circuits and other electrical devices.

### CROCKER, WAGNER ASSUME NEW DUTIES

Sabin Crocker now holds the new position of mechanical engineering consultant in The Detroit Edison Co.'s system engineering department, and will report in a staff capacity to Samuel M. Dean, chief engineer of the system, to undertake special assignments in the field of system engineering. Since graduating from the University of Michigan in 1915, Mr. Crocker has been continuously engaged with Detroit Edison with the exception of two years during World War I. He is well known in the piping field through serving on ASTM and ASA committees concerned with piping codes, specifications, and standards, is the author of several ASME papers and of the Piping Handbook, has been a frequent contributor to Heating, Piping & Air Conditioning, and is a member of its board of consulting and contributing editors.

Mr. Crocker

Harvey A. Wagner, author of several articles in HPAC, has been appointed senior engineer-mechanical, engineering division of Detroit Edison. Mr. Wagner graduated from the University of Michigan in 1927, and except for one year immediately after graduation, has been employed in Detroit Edison's engineering division. He is the author of a number of articles on high pressure, high temperature piping design, represents the company or the power industry on several ASA and ASTM committees for piping materials, standards, and codes.

Mr. Wagner

### E. M. FLANNERY HEADS REMA

E. M. Flannery, vice president of the Bush Mfg. Co., was elected president of the Refrigeration Equipment Manufacturers Association at the annual meeting in Chicago, April 9 to 11. He succeeds H. F. Spoehrer, vice president of the Sporlan Valve Co., who becomes a member of the association advisory board.

### ROBERT H. WHITELEY DIES SUDDENLY

Robert H. Whiteley, president of the Randall Graphite Products Corp., Chicago, died of a heart attack April 25. Mr. Whiteley, who was 57, was one of the first members of the firm, having become associated with it 38 years ago.

He is survived by three sons in the business—Robert L., Howard C., and Kenneth E.—and by his widow and a sister.

# Heating Piping & Air Conditioning

with which is merged the Aerologist

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## Journal Section of the American Society of Heating and Ventilating Engineers


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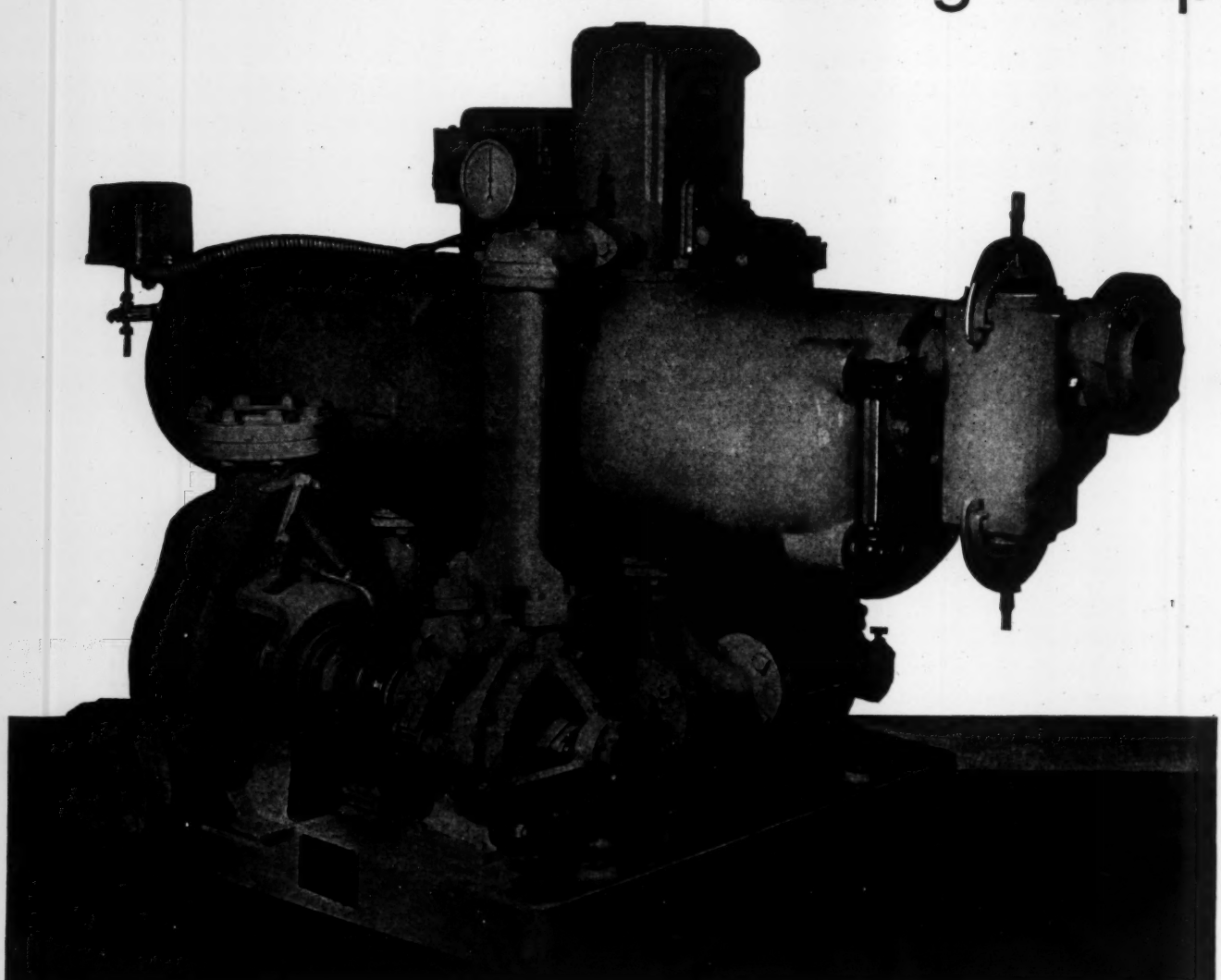
# Where Heat Must Not Fail-- Install this SAFE Heating Pump!



In Hospitals, Greenhouses, Schools, Public Buildings, Theatres, wherever heating systems must not fail, install the Nash Vapor Turbine, for it is entirely independent of electric current failure, and continues to operate as long as there is steam in the system.

More than that, the Vapor Turbine is a most economical pump, for the elimination of electric current does away with current cost, the largest single item in the operation of an ordinary return line heating pump.

Greater savings still are effected by the Vapor Turbine in the system, for the reason that this pump operates continuously. It is the only pump that can do this with economy. Continuous operation means uniform circulation, and uniform circulation saves steam.

The Nash Vapor Turbine has but one moving part, rotating in the casing without metallic contact, and requiring no internal lubrication. Quiet, compact, and trouble-proof. Bulletin A-290 is free on request.

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*Methods of Making*

# PIPING FLEXIBILITY

*Analyses*

## *GENERAL ANALYTICAL METHOD*

**By H. V. Wallstrom, Mechanical Engineer**

**The M. W. Kellogg Co.**

**THERE ARE numerous methods of analyzing the flexibility of piping systems and a great amount of material on this important subject has been published. In the June 1946 HPAC, Sabin Crocker and Arthur McCutchan reviewed the various methods, and suggested that the advocates of the different schemes each briefly outline the merits of his method and show the solution of two problems; in this way, the reader can judge for himself the suitability of the different calculation procedures in his own piping work. . . . S. W. Spielvogel followed this suggestion in the July issue, explaining the essentials of the elastic center method, and in September M. J. Fish discussed the graphical method. In November, W. G. Hooper covered the tabular integration method and solved the two problems given by Messrs. Crocker and McCutchan by that means. In the January 1947 number, R. C. DeHart explained the moment distribution method and this month the general analytical* method is considered.**

DURING THE EARLIER stages of development of piping flexibility analysis, W. H. Shipman** presented an attractive analytical solution for single plane systems in which he introduced pre-integrated shape coefficients, a feature now incorporated in all similar methods. G. W. Watts and W. R. Burrows of the Standard Oil Co. (Indiana)*** enlarged this solution to cover the three plane problem with two fixed ends by their ingenious application of cyclic permutation of the coordinates.

Recognizing the advantages of this completely analytical development, The M. W. Kellogg Co. engineers extended the fundamental theory to cover the general problem involving pipe lines of several branches and various types of intermediate and end restraints. Thus it is now possible to solve complex systems with the same simplicity as obtains for a line with two points of fixation; the complexity is limited only by the impracticability at the present time of solving more than about 18 equations representing four points of end fixation for a multiplane system or seven points for a single plane system. During recent years a further extension has been added to include concentrated and uniform loading, such as produced by weight and wind. This versatility suggests the title, the General Analytical Method.*

### Practical Application Only Discussed

In this article only the practical application of this method is discussed; the reader is referred to the publication Design of Piping Systems (The M. W. Kellogg Co., New York, 1941) for the development and derivation of the various formulas involved. Briefly, the method is this: The pre-integrated terms involved in the solution are derived from the work equation. Summation of these terms may be made so as to represent an integration over any given system yielding the coefficients of a set of simultaneous linear equations involving the unknown forces and moments. The solution gives forces and moments at the origin but the moments existing at any point in the system may be found by a simple transformation. Before discussing the general procedure for making a piping flexibility analysis, a brief

*In the introductory article of this series on Methods of Making Piping Flexibility Analyses, published in the June 1946 HPAC, the methods of W. H. Shipman, Standard Oil Co. (Indiana) and The M. W. Kellogg Company were listed under the heading "Shipman's Method."

**Design of Piping to Care for Expansion, by W. H. Shipman, ASME Transactions Vol. 51, 1929.

***Standard Oil Co. (Indiana) Company Report, Piping Design, by G. W. Watts and W. R. Burrows, 1932.

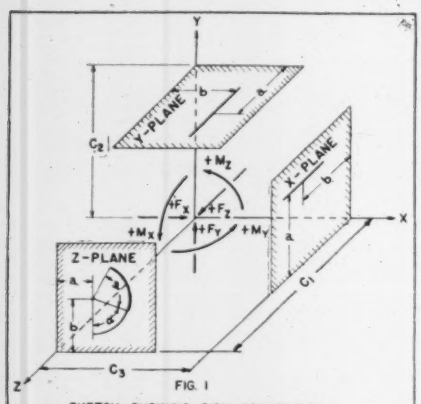



FIG. 1

SKETCH SHOWING SIGN CONVENTION AND ORIENTATION OF WORKING PLANES IN RESPECT TO COORDINATE AXES

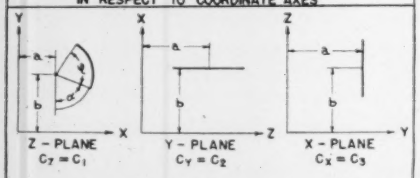



FIG. 2

SKETCHES OF WORKING PLANES ORIENTED IN RELATION TO GENERAL FORMULAS

description of certain essential concepts is presented.

*Sign Convention*—The tri-axial coordinate system shown in Fig. 1 is used. The positive direction of the *X*, *Y*, and *Z* axes is always to the right, upward and toward the observer respectively. Forces are considered positive when acting in the positive direction of the axes. Signs of the moments are established in accordance with the right hand rule relation to the direction of the axes.

*Members*—The simplicity of the resulting expressions (presented on the form sheets) is to a large extent the result of subdividing the piping system into its component parts—such as straight lengths, quarter-circular bends, etc. Hence, each member—whether straight or curved—can be assigned to a plane compatible with its location in the line, called its working plane.

*Working Planes*—The working planes are designated by the coordinate axes perpendicular to these planes and collectively contain all the members of a piping system. Each member is assigned to one working plane only, the choice of plane being dependent on adjacent members. Fig. 1 shows various members in space and the working planes to which they are assigned. Fig. 2 shows the reorientation of the working planes necessary for defining the positions of the various members for calculation. In each case, the horizontal distance from a vertical axis to the center of a straight member or to the center of radius of a circular member is designated as $a$ while the corresponding vertical distance is designated as $b$. The values $c_x$, $c_y$, and $c_z$ are the distances of the working planes from the origin.

*Shape Coefficients*—The shape coefficients are calculated from pre-integrated expressions corresponding to elements of shape for the various members. For a single plane line there are six coefficients to be calculated for each member; for a multiplane line a maximum of six additional are required. The formulas for the shape coefficients of a straight member consisting of plain or corrugated pipe located in a general position are given in the first column of Fig. 3. In the remaining columns the formulas for a plain member in the horizontal or vertical positions are given in a form convenient for calculation. The formulas for the shape coefficients of a circular member are somewhat more elaborate and are presented on the form sheet Fig. 4 which includes the calculation of member *1-2* for the Hovgaard example bend described later. In the lower right hand corner certain trigonometric constants are tabulated covering quarter-circular members in various positions. These constants or the constants for any arc in any position are computed from the following general formulas:

$\phi$ = angle in radians

$c_a = \cos\ \alpha - \cos\ (\alpha + \phi)$

$c_b = \sin\ \alpha - \sin\ (\alpha + \phi)$

$c_{ab} = 0.25\ [\cos\ 2\ (\alpha+\phi) - \cos\ 2\alpha]$

$c_{aa} = 0.5\phi - 0.25\ [\sin\ 2\ (\alpha+\phi) - \sin\ 2\alpha]$

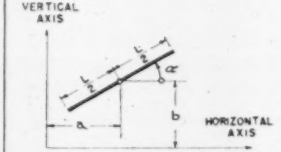
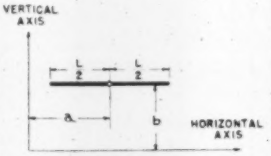
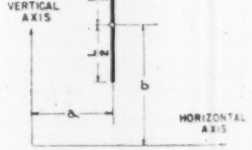
$c_{bb} = 0.5\phi + 0.25\ [\sin\ 2\ (\alpha + \phi) - \sin\ 2\alpha]$

[Note: The angles $\alpha$ and $\phi$ are defined in Fig. 2.]

The symbol $K$ in Figs. 4, 7, and 12 represents both the Karman and the Vigness flexibility factors as applied to circular members. The same symbol in Fig. 3 for straight members represents a flexibility factor applied where the pipe is corrugated.

### Diagonal Symmetry and the Equations

Proper combination and arrangement of the shape constants results in a symmetrical array of coefficients of a system of simultaneous linear equations involving the unknown forces and moments. Equations exist for each component displacement, rotational or translatory, which are called the moment or force equations respectively. If the moments and forces are written in a given order in

| GENERAL POSITION | HORIZONTAL MEMBER | VERTICAL MEMBER |
|---|---|---|
|  |  |  |
| K = 1 FOR PLAIN PIPE<br>K = 5 FOR CORRUGATED PIPE | PLAIN PIPE | PLAIN PIPE |
| $S = QKL$ | $S = Q \cdot L$ | $S = QL$ |
| $S_a = S \cdot a$ | $S_a = S \cdot a$ | $S_a = S \cdot a$ |
| $S_b = S \cdot b$ | $S_b = S \cdot b$ | $S_b = S \cdot b$ |
| $S_{ab} = S_a \cdot b + QK\ L^3\ 1/12\ \sin\alpha \cos\alpha$ | $S_{ab} = S_b \cdot a$ | $S_{ab} = S_a\ b$ |
| $S_{aa} = S_a \cdot a + QK\ L^3\ 1/12\ \cos^2\alpha$ | $S_{aa} = S_a \cdot a + Q\ L^3\ 1/12$ | $S_{aa} = S_a \cdot a$ |
| $S_{bb} = S_b \cdot b + QK\ L^3\ 1/12\ \sin^2\alpha$ | $S_{bb} = S_b \cdot b$ | $S_{bb} = S_b\ b + Q\ L^3\ 1/12$ |
| $U = QL\ (K \cos^2\alpha + 1.3 \sin^2\alpha)$ | $U = S$ | $U = 1.3S$ |
| $V = QL\ (K \sin^2\alpha + 1.3 \cos^2\alpha)$ | $V = 1.3S$ | $V = S$ |
| $W = QL\ (1.3 - K) \sin\alpha \cos\alpha$ | $W = 0$ | $W = 0$ |
| $U_o = U \cdot a - W \cdot b$ | $U_o = S_a$ | $U_o = 1.3S_A$ |
| $V_o = V \cdot b - W \cdot a$ | $V_o = 1.3S_b$ | $V_o = S_b$ |
| $W_o = U_o a + V_o b + QK\ L^3\ 1/12$ | $W_o = S_{aa} + 1.3S_{bb}$ | $W_o = S_{bb} + 1.3S_{aa}$ |

$Q = \frac{EI}{E_N I_N}$ = CORRECTION FACTOR APPLICABLE TO MEMBERS OF A DIFFERENT RIGIDITY $E_N I_N$ FROM THAT OF THE MAJORITY

FIG. 3 — SHAPE COEFFICIENTS FOR STRAIGHT MEMBERS

| FUNCTIONS OF RADIUS | | | TRIGONOMETRIC CONSTANTS | | MEMBER 1–2 HOVGAARD BEND | COORDINATES | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| K | | 2.29 | α | 180° | ⓐ | − | 1.49 | ⓑ | − | 3.02 | ab | + | 4.50 | $a^2$ | + | 2.22 | $b^2$ | + | 9.12 |
| 1 | | 1.00 | φ | 90° | 2a | − | 2.98 | 2b | − | 6.04 | 2ab | + | 9.00 | | | | | | |
| R | | 3.02 | | | CALCULATION OF SHAPE COEFFICIENTS | | | | | | | | | | | | | | |
| KQR | + | 6.92 | φ | +1.5708 | S = + 10.87 | + S·a | − | 16.20 | + S·b | − | 32.83 | + S·ab | + | 48.92 | + S·$a^2$ | + | 24.13 | + S·$b^2$ | + | 99.13 |
| $KQR^2$ | + | 20.90 | $C_a$ | −1.0000 | | + $S'_a$ | − | 20.90 | | | | + $S'_a$·b | + | 63.12 | + $S'_a$·2a | + | 62.28 | | | |
| DITTO | | DITTO | $C_b$ | +1.0000 | | | | | + $S'_b$ | + | 20.90 | + $S'_b$·a | − | 31.14 | | | | + $S'_b$·2b | − | 126.24 |
| $KQR^3$ | + | 63.12 | $C_{ab}$ | −.5000 | | | | | | | | + $S'_{ab}$ | − | 31.56 | | | | | | |
| DITTO | | DITTO | $C_{aa}$ | +.7854 | | | | | | | | | | | + $S'_{aa}$ | + | 49.57 | | | |
| DITTO | | DITTO | $C_{bb}$ | +.7854 | | | | | | | | | | | | | | + $S'_{bb}$ | + | 49.57 |
| | | | | | S = + 10.87 | $S_a$ = − | 37.10 | $S_b$ = − | 11.93 | | | $S_{ab}$ = + | | 49.34 | $S_{aa}$ + | | 135.98 | $S_{bb}$ = + | | 22.46 |
| QR | + | 3.02 | $KC_{bb}+1.3C_{aa}$ | +2.8196 | u = + 8.52 | + u·a | − | 12.69 | | | | + u·$a^2$ | + | 18.91 | φ = 90° = + 1.5708 | | | | | |
| DITTO | | DITTO | $KC_{aa}+1.3C_{bb}$ | +2.8196 | V = + 8.52 | | | | + V·b | − | 25.73 | + V·$b^2$ | + | 77.70 | α | 0° | 90° | 180° | | 270° |
| DITTO | | DITTO | $(K-1.3)C_{ab}$ | −.4950 | W = − 1.49 | − W·b | − | 4.50 | − W·a | − | 2.22 | −W·2ab | + | 13.41 | $C_a$ | +1 | +1 | −1 | | −1 |
| $QR^2$ | + | 9.12 | $1.3C_a$ | −1.3000 | | + $U'_o$ | − | 11.86 | | | | + $U'_o$·2a | + | 35.34 | $C_b$ | −1 | +1 | +1 | | −1 |
| DITTO | | DITTO | $1.3C_b$ | +1.3000 | | | | | + $V'_o$ | + | 11.86 | + $V'_o$·2b | − | 71.63 | $C_{ab}$ | −.5 | +.5 | −.5 | | +.5 |
| $QR^3$ | + | 27.54 | $1.3\varphi$ | +2.0420 | | | | | | | | + $W'_o$ | + | 56.24 | $C_{aa} = C_{bb}$ = +.7854 | | | | | |
| | | | | | | $U_o$ = − | 29.05 | $V_o$ = − | 16.09 | | | $W_o$ = + | | 129.97 | 1.3φ = 2.0420 | | | | | |
| SUMMATION MADE WITH W = 0 | | | | | | $U_o$ = − | 24.55 | $V_o$ = − | 13.87 | | | $W_o$ = + | | 116.56 | | | | | | |

FIG. 4
FORM SHEET FOR SHAPE COEFFICIENTS FOR CIRCULAR MEMBERS

each equation and the equations corresponding to the moments and forces are written in the same order vertically, the determinant formed by the coefficients will be symmetrical about a principal diagonal. Accordingly, in setting up the equations, only the values on and above this diagonal need be written.

While the solution of a large number of simultaneous equations involves many time-consuming operations, these operations are minimized if, in the solution, equations are eliminated in such a way that symmetry is retained in the remaining equations. Fig. 5 illustrates how the $M_z$ term and the $M_z$ equation may be eliminated from the equations for a single Z-plane bend, thus reducing the system from three (shown on lines 1, 3, and 6) to two (shown on lines 5 and 8). Note that because of the symmetry, the shaded part need not be written. Since symmetry has been preserved, the remaining two equations can be subjected to the same procedure in which the $F_x$ equation is eliminated. The summation resulting in line 8 is then unnecessary, having been made only to show the intermediate group of equations and their symmetry. The above procedure can be applied to any number of equations and although its simplicity is perhaps not immediately apparent, it is a routine readily mastered with practice.

It may be of interest to digress at this point to discuss the use of the elastic center as a means of eliminating the moment equations, thus reducing the size of the linear system to be solved. Referring again to Fig. 5, note that the values which appear in line 2, namely $+(s_a/s)$ and $-(s_b/s)$ are respectively the abscissa (x) and the ordinate (y) of the elastic center. If the origin were located there, the moment equation would be automatically eliminated leaving only the force equations to be solved. However, this involves two coordinate systems, since the three shape coefficients $s$, $s_a$ and $s_b$ must be calculated in an arbitrary coordinate system in order to locate the elastic center while the remaining coefficients $s_{aa}$, $s_{bb}$, and $s_{ab}$ would be calculated in a coordinate system with its origin at the elastic center. Especially if the number of members is large, this procedure is considered more time-consuming and subject to more errors even in single plane problems. For a multiplane bend, an elastic center must be determined for each plane.

### Omitting the Secondary Term

The procedure described in the above-mentioned publication, Design of Piping Systems, is often modified in order to reduce calculating time without introducing appreciable error in the results. This is accomplished by making the secondary term equal to zero but in no other way changing the procedure.

| LINE | EQUATION NO. & OPERATION | $M_{Z0}$ | $F_X$ | $F_Y$ | CONSTANT | = 0 |
|---|---|---|---|---|---|---|
| ① | EQUATION # 1 | $+ S$ | $+ S_b$ | $- S_a$ | 0 | |
| ② | DIVIDE ① BY (−S) | $- 1$ | $-\frac{S_b}{S}$ | $+\frac{S_a}{S}$ | 0 | |
| | | | | | | |
| ③ | EQUATION # 2 | $+ S_b$ | $+ S_{bb}$ | $- S_{ab}$ | $-EI\,\Delta_x$ | |
| ④ | MULTIPLY EQ. # 1 [LINE ①] BY $(-S_b/S)$ [FROM LINE ②] | $- S_b$ | $-\frac{(S_b)^2}{S}$ | $+\frac{S_a S_b}{S}$ | 0 | |
| ⑤ | ADD EQUATION # 2 [LINE ③] AND EQUATION [LINE ④] | 0 | $+ S_{bb} - \frac{(S_b)^2}{S}$ | $-\left(S_{ab} - \frac{S_a S_b}{S}\right)$ | $-EI\,\Delta_x$ | |
| | | | | | | |
| ⑥ | EQUATION # 3 | $- S_a$ | $- S_{ab}$ | $+ S_{aa}$ | $-EI\,\Delta_y$ | |
| ⑦ | MULTIPLY EQ. # 1 [LINE ①] BY $(+S_a/S)$ [FROM LINE ②] | $+ S_a$ | $+\frac{S_a S_b}{S}$ | $-\frac{(S_a)^2}{S}$ | 0 | |
| ⑧ | ADD EQUATION # 3 [LINE ⑥] AND EQUATION [LINE ⑦] | 0 | $-\left(S_{ab} - \frac{S_a S_b}{S}\right)$ | $+ S_{aa} - \frac{(S_a)^2}{S}$ | $-EI\,\Delta_y$ | |

THE THREE GIVEN EQUATIONS # 1, 2, AND 3 ARE REDUCED TO TWO EQUATIONS — LINES ⑤ AND ⑧

FIG. 5 – METHOD OF REDUCING A SYSTEM OF SYMMETRICAL EQUATIONS MAINTAINING THE SYMMETRY

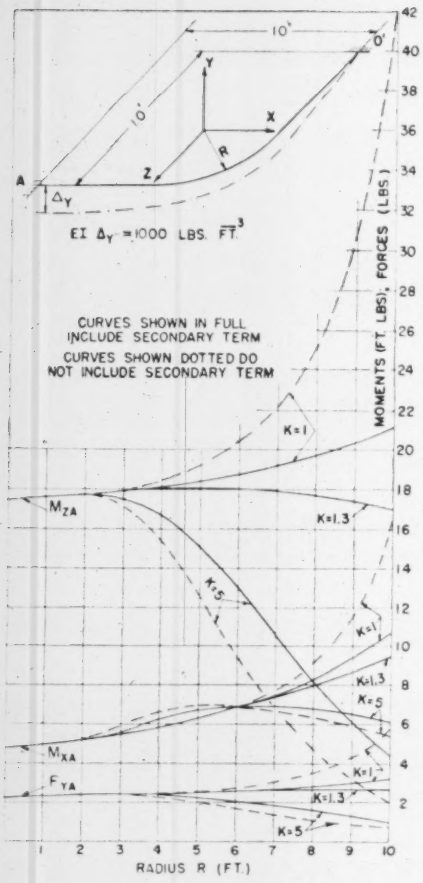
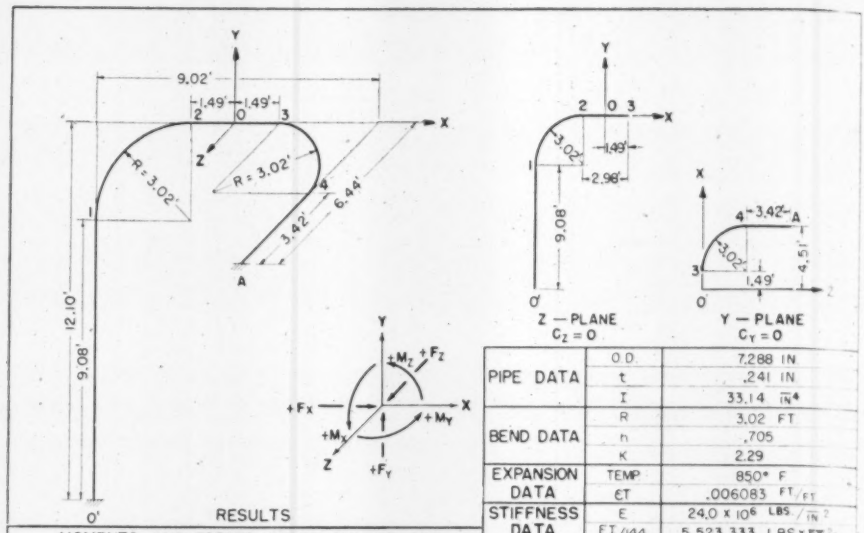



FIG. 6

EFFECT OF THE SECONDARY TERM ON A SYMMETRICAL 90° — BEND WITH VARIOUS LENGTHS OF TANGENTS




RESULTS

MOMENTS AND FORCES ACTING ON ANCHORS

| CASE | W NEGLECTED | | W INCLUDED | |
|---|---|---|---|---|
| POINT | 0' | A | 0' | A |
| $M_X$ FT. LBS | -4,610 | +7,970 | -4,670 | +7,920 |
| $M_Y$ FT. LBS | +1,150 | -6,650 | +1,200 | -6,710 |
| $M_Z$ FT. LBS | +11,090 | -5,450 | +11,090 | -5,280 |
| $F_X$ LBS. | -1,750 | +1,750 | -1,750 | +1,750 |
| $F_Y$ LBS. | -1,720 | +1,720 | -1,710 | +1,710 |
| $F_Z$ LBS. | - 640 | + 640 | - 640 | + 640 |

| | | |
|---|---|---|
| PIPE DATA | O.D. | 7.288 IN. |
| | t | .241 IN. |
| | I | 33.14 $IN^4$ |
| BEND DATA | R | 3.02 FT. |
| | h | .705 |
| | K | 2.29 |
| EXPANSION DATA | TEMP. | 850° F |
| | $\epsilon T$ | .006083 FT./FT. |
| STIFFNESS DATA | E | 24.0 x $10^6$ LBS./$IN^2$ |
| | EI/144 | 5,523,333 LBS x $FT^2$ |
| DISPLACEMENT CALCULATIONS | $\Delta_X$ | - 9.02 x .006083 = = - .05487 FT. |
| | $\Delta_Y$ | - 12.10 x .006083 = = - .07360 FT. |
| | $\Delta_Z$ | - 6.44 x .006083 = = - .03917 FT. |
| | EI$\Delta_X$/144 | - 303,065 LBS. x $FT^3$ |
| | EI$\Delta_Y$/144 | - 406,517 LBS. x $FT^3$ |
| | EI$\Delta_Z$/144 | - 216,349 LBS. x $FT^3$ |

FIG. 7 — ISOMETRIC SKETCH, GIVEN DATA, WORKING PLANES, AND CALCULATED REACTIONS AT ANCHORS — HOVGAARD BEND

The term, identified as the shape coefficient $w$, appears when inclined straight members and circular members are subjected to transverse bending and torsional moments. It is this term which adjusts the shift of the elastic center from the virtual center of gravity. While its magnitude varies with the transverse flexibility factor, $w$ is relatively small compared with the other coefficients. When the transverse flexibility factor is equal to 1.3, it becomes zero. Only in this special case does the elastic center and the virtual center of gravity coincide.

The effect on a particular pipe line of omitting $w$ is illustrated in Fig. 6. A quarter-circular bend with equal tangents is fixed at both ends against rotation but one end is displaced relative to the other in a direction perpendicular to the plane of the line. Values of the moments and forces for the displacement at the end are plotted for various values of the bend radius and for Vigness flexibility factors $K$ = 1, 1.3, and 5. The curves shown full include the term $w$ while the curves shown dotted neglect $w$. Inspection of these curves shows that if the circular member forms a small part of the total line, the discrepancies in omitting $w$ are negligible. However, if the circular member represents the entire pipe line, the errors are of considerable magnitude. For most pipe lines circular members form but a small part of the whole system and the omission of $w$ is therefore justified since it simplifies the solving of the equations and the calculation of the shape coefficients $u_o$, $v_o$, and $w_o$ for inclined straight and circular members. In the elastic center solution the neglect of $w$ is a necessity.

In accordance with present practice, the Hovgaard bend, which will be reviewed in detail, has been calculated by the above modification (except that on Fig. 4 the summations for both cases are given). However, the results obtained with or without the $w$-factor included are tabulated on Fig. 7 for comparison.

## General Procedure for Making a Piping Flexibility Analysis

The complexity of a piping flexibility analysis has been reduced to simple routine calculations which require practically no technical background of the calculator. The use of standardized form sheets is most important for speed and accuracy. The employment of high speed calculating machines is desirable but not essential.

The general procedure for making a piping flexibility analysis is now presented step by step with

| PLANE | Z — PLANE $C_Z$ = 0 | | | Y — PLANE $C_Y$ = 0 | | SUMMATION | PLANE | Z | Y |
|---|---|---|---|---|---|---|---|---|---|
| MEMBER | 0 — 1 | 1 — 2 | 2 — 3 | 3 — 4 | 4 — A | | CU | 0 | 0 |
| SHAPE | | | | | | | CV | 0 | 0 |
| | | | | | | | CW | 0 | 0 |
| $L^3/12$ | 62.38 | | 2.21 | | 3.33 | | $CU_o$ | 0 | 0 |
| L | 9.08 | | 2.98 | | 3.42 | | $C^2U$ | 0 | 0 |
| a | - 4.51 | - 1.49 | 0 | + 3.02 | + 4.73 | | $C^2V$ | 0 | 0 |
| b | - 7.56 | - 3.02 | 0 | + 1.49 | + 4.51 | | $C^2W$ | 0 | 0 |
| S | 9.08 | 10.87 | 2.98 | 10.87 | 3.42 | | S | 22.93 | 14.29 |
| $S_a$ | - 40.95 | -37.10 | 0 | 11.93 | 16.18 | | $S_a$ | - 78.05 | 28.11 |
| $S_b$ | - 68.64 | -11.93 | 0 | 37.10 | 15.42 | | $S_b$ | - 80.57 | 52.52 |
| $S_{ab}$ | 309.58 | 49.34 | 0 | 49.34 | 72.94 | | $S_{ab}$ | 358.92 | 122.28 |
| $S_{aa}$ | 184.68 | 135.98 | 2.21 | 22.46 | 79.86 | | $S_{aa}$ | 322.87 | 102.52 |
| $S_{bb}$ | 581.30 | 22.46 | 0 | 135.98 | 69.54 | | $S_{bb}$ | 603.76 | 205.52 |
| U | 11.80 | 8.52 | 2.98 | 8.52 | 3.42 | | U | 23.30 | 11.94 |
| V | 9.08 | 8.52 | 3.87 | 8.52 | 4.45 | | V | 21.47 | 12.97 |
| W | 0 | 0 | 0 | 0 | 0 | | W | 0 | 0 |
| $U_o$ | - 53.24 | -24.55 | 0 | 13.87 | 16.18 | | $U_o$ | - 77.79 | 30.05 |
| $V_o$ | - 68.64 | -13.87 | 0 | 24.55 | 20.05 | | $V_o$ | - 82.51 | 44.60 |
| $W_o$ | 621.38 | 116.56 | 2.21 | 116.56 | 170.26 | | $W_o$ | 940.15 | 286.82 |

FIG. 8 — CALCULATION OF SHAPE COEFFICIENTS — HOVGAARD BEND

| $M_x$ | | $M_y$ | | $M_z$ | | $F_x$ | | $F_y$ | | $F_z$ | | PLANE |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| +s | | | | | | | | $+s_b$ | | $-s_a$ | | X |
| +u | +11.94 | | 0 | +w | 0 | +cw | 0 | $+u_o$ | + 30.05 | -cu | 0 | Y |
| +v | +21.47 | +w | 0 | | 0 | -cw | 0 | +cv | 0 | $-v_o$ | + 82.51 | Z |
| $A_{xx}$ | +33.41 | $A_{xy}$ | 0 | $A_{xz}$ | 0 | $B_{xx}$ | 0 | $B_{xy}$ | + 30.05 | $B_{xz}$ | + 82.51 | |
| | | +v | | +w | | $-v_o$ | | -cw | | +cv | | X |
| | | +s | + 14.29 | | 0 | $-s_a$ | - 28.11 | | 0 | $+s_b$ | + 52.52 | Y |
| | | +u | + 23.30 | | 0 | -cu | 0 | +cw | 0 | $+u_o$ | - 77.79 | Z |
| | | $A_{yy}$ | + 37.59 | $A_{yz}$ | 0 | $B_{yx}$ | - 28.11 | $B_{yy}$ | 0 | $B_{yz}$ | - 25.27 | |
| | | | | +u | | $+u_o$ | | -cu | | +cw | | X |
| | | | | +v | + 12.97 | +cv | 0 | $-v_o$ | - 44.60 | -cw | 0 | Y |
| | | | | +s | + 22.93 | $+s_b$ | - 80.57 | $-s_a$ | + 78.05 | | 0 | Z |
| | | | | $A_{zz}$ | + 35.90 | $B_{zx}$ | - 80.57 | $B_{zy}$ | + 33.45 | $B_{zz}$ | 0 | |
| | | | | | | $+w_o$ | | $-cu_b$ | | $-cv_o$ | | X |
| | | | | | | $+s_{aa}$ | + 102.32 | $-cv_o$ | 0 | $-s_{ab}$ | - 122.28 | Y |
| | | | | | | $+c^2v$ | 0 | | 0 | $-c^2w$ | 0 | |
| | | | | | | $+s_{bb}$ | + 603.76 | $-s_{ab}$ | - 358.92 | $-cu_b$ | 0 | Z |
| | | | | | | $+c^2u$ | 0 | $-c^2w$ | 0 | | 0 | |
| | | | | | | $C_{xx}$ | + 706.08 | $C_{xy}$ | - 358.92 | $C_{xz}$ | - 122.28 | |
| | | | | | | | | $+s_{bb}$ | | $-s_{ab}$ | | X |
| | | | | | | | | $+c^2u$ | | $-c^2w$ | | |
| | | | | | | | | $+w_o$ | + 286.82 | $-cu_b$ | 0 | Y |
| | | | | | | | | $+s_{aa}$ | + 322.87 | $-cv_o$ | 0 | Z |
| | | | | | | | | $+c^2v$ | 0 | | 0 | |
| | | | | | | | | $C_{yy}$ | + 609.69 | $C_{yz}$ | 0 | |
| | | | | | | | | | | $+s_{aa}$ | | X |
| | | | | | | | | | | $+c^2v$ | | |
| | | | | | | | | | | $+s_{bb}$ | + 205.52 | Y |
| | | | | | | | | | | $+c^2u$ | 0 | |
| | | | | | | | | | | $+w_o$ | + 940.15 | Z |
| | | | | | | | | | | $C_{zz}$ | +1145.67 | |

FIG. 9 — PLANE SUMMATION OF SHAPE COEFFICIENTS HOVGAARD BEND

calculations, using for the example the three-dimensional bend given in the introductory article in the June 1946 HPAC. This bend, when originally analyzed by Hovgaard, included extraneous movements which, since they have been neglected in the introductory article, are not considered herein.

1) All data necessary for the analysis, including a dimensioned sketch of the line, are entered on the form sheet Fig. 7. Note that calculations are based on dimensions in feet.

2) The component displacements are entered as $\Delta_x$, $\Delta_y$, and $\Delta_z$. For the purpose of calculating these displacements, the line is visualized as a cantilever with the end $0'$ considered fixed and the other end $A$ permitted to expand freely. The magnitudes of the component displacements are the translatory movements required to return the free end to its original position. The signs of these displacements are determined by their directions with respect to the coordinate axes.

3) The location of the origin for the coordinate system is selected in such a way that the calculation of the shape coefficients is simplified to the greatest possible extent. In this example it is located so that both the $c_z$ and $c_y$ distances are zero, and so that certain constants are duplicated in the calculations.

4) The members are shown in their respective working planes. With practice, this operation may be performed mentally.

5) A sketch of the shape of each member and its $a$ and $b$ distances are entered on the form sheet Fig. 8, whereon the shape coefficients are calculated as described above. The shape coefficients for the members in each plane are summed up and entered in the columns at the lower right and the products of certain coefficients with the $c$ distances ($c_y = c_z = 0$ for this problem) are entered in the columns above.

6) These summations are transferred to the form sheet Fig. 9 according to the plane indicated in the column at the extreme right. In this example no members are calculated in the X-plane, hence no entries are made for that plane. Groups of these coefficients are added as indicated to obtain the values labelled $A_{xx}$, $A_{xy}$, etc.

7) The values $A_{xx}$, $A_{xy}$, etc., which are the coefficients of a system of linear equations, are transferred from form sheet Fig. 9 to lines 1 to 6 inclusive on form sheet Fig. 10. Also, the constants ($EI\Delta_x/144$), ($EI\Delta_y/144$), and ($EI\Delta_z/144$) from Fig. 7 are entered with opposite signs on Fig. 10 in equations 4, 5 and 6 in the column headed "Constants" (the constants for the moment equation are zero since the ends of the bend are fixed). The equations are then solved in order to obtain the forces

| CHECK | $M_{xo}$ | $M_{yo}$ | $M_{zo}$ | $F_x$ | $F_y$ | $F_z$ | CONSTANTS |
|---|---|---|---|---|---|---|---|
| 1 | +104,578 | 0 | 0 | 0 | - 51,814 | - 52,763 | +1 |
| 2 | 0 | - 65,373 | 0 | + 49,213 | 0 | + 16,160 | 0 |
| 3 | 0 | 0 | - 83,379 | + 141,056 | - 57,677 | 0 | 0 |
| 4 | 0 | + 48,886 | +187,126 | - 1,236,148 | + 618,875 | + 78,196 | - 303,065 |
| 5 | + 94,060 | 0 | - 77,689 | + 628,368 | - 1051,270 | 0 | - 406,531 |
| 6 | + 258,267 | + 43,947 | 0 | + 214,078 | 0 | - 732,633 | - 216,341 |

| | $M_{xo}$ | $M_{yo}$ | $M_{zo}$ | $F_x$ | $F_y$ | $F_z$ | CONSTANTS |
|---|---|---|---|---|---|---|---|
| 1 | + 33.41 | 0 | 0 | 0 | + 30.05 | + 82.51 | 0 |
| | - 1.00000 | 0 | 0 | 0 | - .899 43 | - 2.469 62 | 0 |
| $M_{xo}$ = + 3 130. 13 = | | 0 | 0 | 0 | + 1.550.86 | + 1579.27 | 0 |
| 2 | + 37.59 | 0 | 0 | - 28.11 | 0 | - 25.27 | 0 |
| | | 0 | 0 | 0 | 0 | 0 | 0 |
| Σ | + 37.59 | 0 | 0 | - 28.11 | 0 | - 25.27 | 0 |
| | - 1.00000 | 0 | 0 | + .747 81 | 0 | + .672 25 | 0 |
| $M_{yo}$ = - 1739.10 = | | 0 | 0 | - 1 309.21 | 0 | - 429.89 | 0 |
| 3 | + 35.90 | | 0 | - 80.57 | + 33.45 | 0 | 0 |
| | | | 0 | 0 | 0 | 0 | 0 |
| | | | 0 | 0 | 0 | 0 | 0 |
| Σ | + 35.90 | | 0 | - 80.57 | + 33.45 | 0 | 0 |
| | - 1.00000 | | 0 | + 2.244 29 | - .931 75 | 0 | 0 |
| $M_{zo}$ = - 2 322.53 = | | | - 3 929.12 | + 1 606.59 | 0 | 0 | |
| 4 | + 706.08 | | | - 358.92 | - 122.28 | + 303.065 | |
| | 0 | | | 0 | 0 | | |
| | - 21.02 | | | 0 | - 18.90 | | |
| | - 180.82 | | | + 75.07 | 0 | 0 | |
| Σ | + 504.24 | | | - 283.85 | - 141.18 | + 303.065 | |
| | - 1.00000 | | | + .562 93 | + .279 99 | - 601.03 | |
| $F_x$ = - 1 750.72 = | | | | - 970.64 | - 179.05 | - 601.03 | |
| 5 | + 609.69 | | | | 0 | + 406.517 | |
| | - 27.03 | | | | - 74.21 | 0 | |
| | 0 | | | | 0 | 0 | |
| | - 31.17 | | | | 0 | 0 | |
| | - 159.79 | | | | - 79.47 | + 170.604 | |
| Σ | + 391.70 | | | | - 153.68 | + 577.121 | |
| | - 1.00000 | | | | + .392 34 | - 1473.38 | |
| $F_y$ = - 1 724.27 = | | | | | - 250.89 | - 1 473.38 | |
| 6 | + 1 145.67 | | | | + 1 216.349 | | |
| | - 203.77 | | | | 0 | | |
| | - 16.99 | | | | 0 | | |
| | 0 | | | | 0 | | |
| | - 39.53 | | | | + 84.855 | | |
| | - 60.29 | | | | + 226.428 | | |
| Σ | + 825.09 | | | | + 527.632 | | |
| | - 1.00000 | | | | - 639.48 | | |
| $F_z$ = - 633.48 | | | | | | | |

FIG. 10 — SIMULTANEOUS EQUATIONS — HOVGAARD BEND

| POINT | 0' | 1 | 2 | 3 | 4 | A |
|---|---|---|---|---|---|---|
| X | - 4.51 | ⟶ | -1.49 | +1.49 | +4.51 | ⟶ |
| Y | -12.10 | -3.02 | 0 | 0 | 0 | 0 |
| Z | 0 | 0 | 0 | 0 | +3.02 | +6.44 |
| $+M_{X0}$ | + 3,130 | ⟶ | ⟶ | ⟶ | ⟶ | ⟶ |
| $+F_Y Z$ | 0 | 0 | 0 | 0 | - 5,207 | -11,104 |
| $-F_Z Y$ | - 7,738 | - 1,931 | 0 | 0 | 0 | 0 |
| $+M_X$ | - 4,608 | + 1,199 | + 3,130 | ⟶ | - 2,077 | - 7,974 |
| $+M_{Y0}$ | - 1,739 | ⟶ | ⟶ | ⟶ | ⟶ | ⟶ |
| $+F_Z X$ | + 2,884 | ⟶ | + 953 | - 953 | - 2,884 | ⟶ |
| $-F_X Z$ | 0 | 0 | 0 | 0 | + 5,287 | +11,275 |
| $+M_Y$ | + 1,145 | ⟶ | - 786 | - 2,692 | + 664 | + 6,652 |
| $+M_{Z0}$ | - 2,323 | ⟶ | ⟶ | ⟶ | ⟶ | ⟶ |
| $+F_X Y$ | +21,184 | + 5,287 | 0 | 0 | 0 | 0 |
| $-F_Y X$ | - 7,776 | ⟶ | - 2,569 | + 2,569 | + 7,776 | ⟶ |
| $+M_Z$ | +11,085 | - 4,812 | - 4,892 | + 246 | + 5,453 | ⟶ |

FIG. 11 — CALCULATION OF MOMENTS AT VARIOUS POINTS HOVGAARD BEND

and moments at the origin. At the top of form sheet Fig. 10 the solution has been substituted in each equation to obtain values of the constants which (with signs changed) check closely with the original constants below. This indicates the solution to be correct.

8) The moments at various points are calculated on the form sheet Fig. 11 for the purpose of locating the point of maximum stress. However, the stress calculation, while one of the main objects of a flexibility analysis, is here omitted so as to confine the issue to the main topic of flexibility. The moments and forces acting on the anchor points have been transferred back to form sheet Fig. 7 so as to present the results together with the statement of the problem.

### Single Plane Flexibility Analysis

For a single plane bend the general procedure of a piping flexibility analysis is the same as for a multiplane bend except that a considerable number of operations are not required. Fig. 12 shows the complete analysis of the Z-bend suggested as the other example in the introductory article published in the June 1946 HPAC.

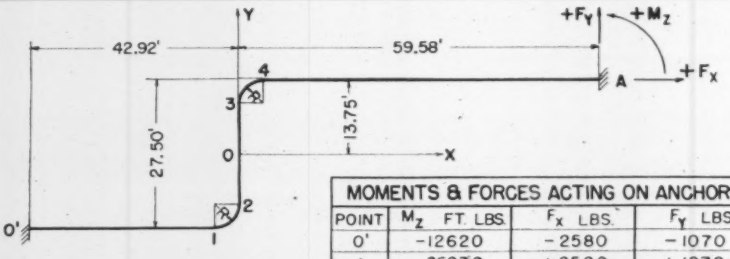
## H. K. IHRIG RECEIVES METALLURGICAL AWARD

Dr. Harry K. Ihrig, director of laboratories of the Globe Steel Tubes Co., was presented the Robert W. Hunt Award at the annual banquet of the American Institute of Mining and Metallurgical Engineers for his paper on the effect of various elements on the hot workability of steels.

The paper reported that a quantitative, hot workability test has been devised, and gives the effects of various elements in steel on this property. Manganese, nickel, and chromium (above 9 per cent) were found beneficial, whereas sulphur, selenium, silicon, nitrogen, molybdenum, columbium, lead, tin, and chromium (below 9 per cent) had a detrimental effect on hot workability. Oxygen, carbon, phosphorus, cobalt, vanadium, and titanium were reported to have little or no effect.

## HEWITT NAMED GENERAL MANAGER

Announcement of the appointment of C. L. Hewitt, Jr., as general sales manager was recently made by Harold P. Mueller, president of the L. J. Mueller Furnace Co.




MOMENTS & FORCES ACTING ON ANCHORS

| POINT | $M_Z$ FT. LBS. | $F_X$ LBS. | $F_Y$ LBS. |
|---|---|---|---|
| 0' | -12620 | - 2580 | - 1070 |
| A | -25930 | + 2580 | + 1070 |

CALCULATION OF SHAPE COEFFICIENTS

| MEMBER | 0'—1 | 1—2 | 2—3 | 3—4 | 4—A | 0'—A |
|---|---|---|---|---|---|---|
| SHAPE | — | ⌐ | \| | ⌐ | — | Σ |
| $L^3/12$ | 6,029.61 | | 1,302.08 | | 16,538.45 | |
| L | 41.67 | | 25.00 | | 58.33 | |
| a | - 22.09 | - 1.25 | 0 | + 1.25 | + 30.42 | |
| b | - 13.75 | - 12.50 | 0 | + 12.50 | + 13.75 | |
| S | 41.67 | 9.25 | 25.00 | 9.25 | 58.33 | 143.50 |
| $S_a$ | - 920.49 | - 4.20 | 0 | 4.20 | 1,774.40 | 853.91 |
| $S_b$ | - 572.96 | -122.99 | 0 | 122.99 | 802.04 | 229.08 |
| $S_{ab}$ | 12,656.69 | 57.18 | 0 | 57.18 | 24,398.06 | 37,169.11 |
| $S_{aa}$ | 26,363.23 | 3.26 | 0 | 3.26 | 70,515.70 | 96,885.45 |
| $S_{bb}$ | 7,878.20 | 1,636.54 | 1,302.08 | 1,636.54 | 11,028.05 | 23,481.41 |

CALCULATION OF MOMENT AT VARIOUS POINTS

| POINT | 0' | 1 | 2 | 3 | 4 | A |
|---|---|---|---|---|---|---|
| X | - 42.92 | - 1.25 | 0 | 0 | + 1.25 | + 59.58 |
| Y | - 13.75 | ⟶ | - 12.50 | + 12.50 | + 13.75 | ⟶ |
| $M_{Z0}$ | - 2238 | ⟶ | ⟶ | ⟶ | ⟶ | ⟶ |
| $+F_X \cdot Y$ | + 35432 | ⟶ | +32211 | -32211 | - 35432 | ⟶ |
| $-F_Y \cdot X$ | - 45813 | - 1334 | 0 | 0 | + 1334 | +63596 |
| $M_Z$ | - 12619 | +31860 | +29973 | -34449 | - 36336 | +25926 |

| | | |
|---|---|---|
| PIPE DATA | O.D. | 10.75 IN. |
| | t | .593 IN. |
| | I | 244.8 $IN.^4$ |
| BEND DATA | R | 1.25 FT |
| | h | .345 |
| | K | 4.71 |
| EXPANSION DATA | TEMP. | 725° F. |
| | $\epsilon T$ | .00490 FT/FT |
| STIFFNESS DATA | E | 25.0 x $10^6$ LBS/$IN^2$ |
| | EI/144 | 42,500,000 LBS. $FT^2$ |
| DISPLACEMENT CALCULATIONS | $\Delta_X$ | -102.50 x .00490 = -.50225 FT. |
| | $\Delta_Y$ | - 27.50 x .00490 = - .13475 FT. |
| | $EI\Delta_X/144$ | - 21,345,625 LBS. $FT^3$ |
| | $EI\Delta_Y/144$ | - 5,726,875 LBS. $FT^3$ |

EQUATIONS

| | | $M_{Z0}$ | $F_X$ | $F_Y$ | CONSTANTS |
|---|---|---|---|---|---|
| CHECK | | - 321.167 | - 590,303 | + 911.472 | +2 |
| | | -512,704 | -60,507,837 | +39,674,680 | -21,345,861 |
| | | +1,911.136 | +95,778,849 | -103,416,498 | - 5,726,513 |
| | | $M_{Z0}$ | $F_X$ | $F_Y$ | CONSTANTS |
| 1 | | + 143.50 | + 229.08 | - 853.91 | 0 |
| | | -1 | ①-1.59638 | ②+5.95059 | SEE NOTE |
| $M_{Z0}$ | -2238.10 = | | + 4,113.62 | - 6,351.72 | 0 |
| | 2 | | +23,481.41 | - 37,169.11 | + 21,345,625 |
| | | | - 365.70 | + 1,363.16 | 0 |
| | Σ | | +23,115.71 | - 35,805.95 | + 21,345,625 |
| | | | -1 | + 1.54899 | |
| | $F_X$ | | - 2,576.84 = | - 1,653.41 | - 923.43 |
| | 3 | | | + 96,885.45 | + 5,726,875 |
| | | | | - 5,081.27 | 0 |
| | | | | - 55,463.06 | + 33,064,160 |
| | Σ | | | + 36,341.12 | + 38,791,035 |
| | $F_Y$ | | | - 1,067.41 | |

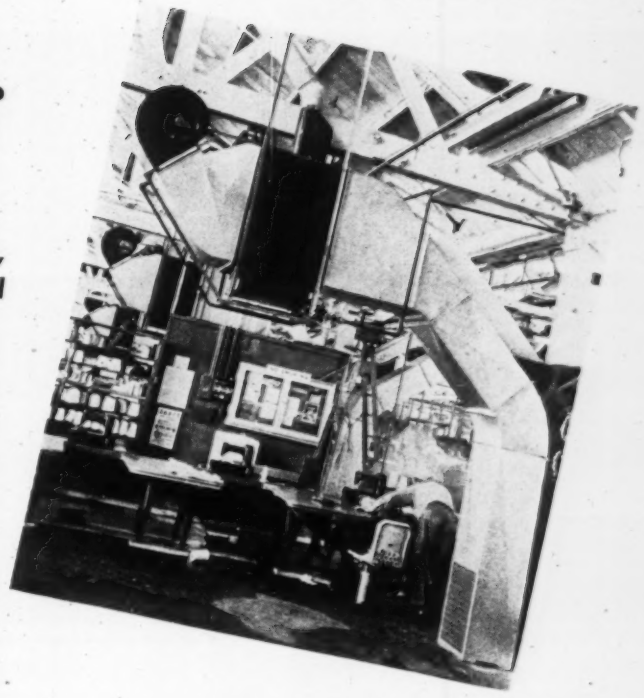
NOTE:-
ELASTIC CENTER COORDINATES:
①Y = + 1.59638
②x = +5.95059

FIG. 12 — Z — BEND WITH WELDING ELBOWS

# Getting Heat Down to the Floor

***O. W. Ott, Consulting Mechanical Engineer, Tells How Bag Plant Problem Was Solved***

Each suspended unit serves an area about 20 ft wide and 50 ft long



IN CALIFORNIA, the confirmed boosters of our wonderful climate sometimes overplay their hands and an eastern industry will establish a branch in Los Angeles and forget all about the heating until they have moved in and Old Man Winter comes around the corner. Then they send out an SOS to the heating engineer.

Such was the case with the Bemis Bag Co. plant in East Los Angeles, a building 140 ft wide by 350 ft long with corrugated iron walls and roof, dormer windows above the truss crowns, and concrete floor on dirt—first class heating losses and very cold floors in our winter months. The production line includes rows of sewing machines and other bag making equipment operated by girls seated on chairs or standing directly on the concrete floor. The center 160 ft, lengthwise, of the building is the working area and the two ends are used for storage of materials and finished product.

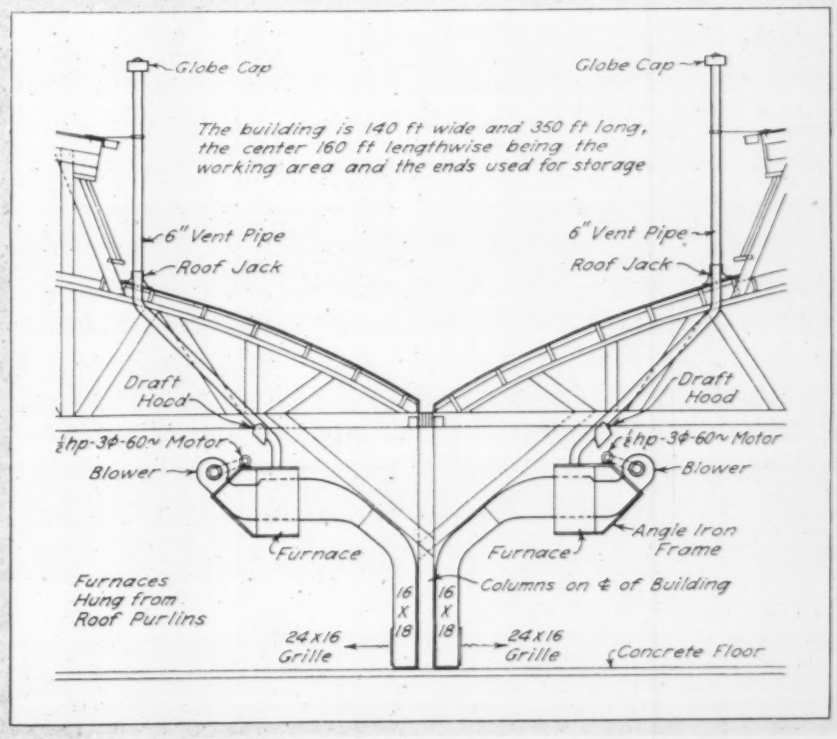
The problem was how to get heat on the floor where the girls work and not heat the entire building.

The problem was solved by bringing the warm air down to the floor; the installation is proving quite satisfactory.

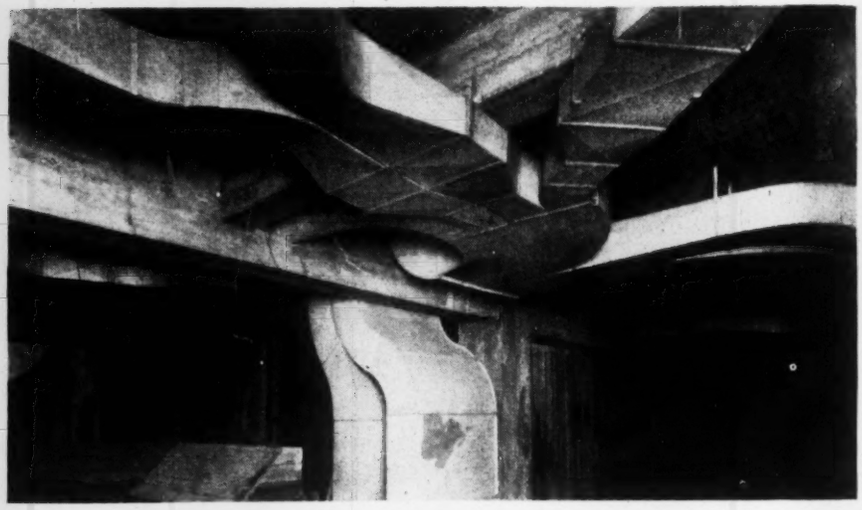
Duct type gas furnaces with integrally mounted centrifugal fans were suspended below the roof trusses and warm air ducts were brought down the diagonal braces and the faces of the columns to the floor on each side of the center row of columns. Each furnace serves an area about 20 ft wide by 50 ft long occupied by the rows of sewing or bag machines.

The fans deliver warm air at 120 deg and approximately 1300 fpm velocity and operate on a recirculation basis. Each unit is rated at 175,000 Btu input and 140,000 Btu output.

**HOW TO HEAT the area where the girls work in a bag company plant in Los Angeles without heating the entire building was the problem, and the answer was found to be duct type gas furnaces suspended below the roof trusses with warm air ducts brought down the diagonal braces and columns to the floor on each side of the center row of columns**



# Slide Rule Is All That's Needed TO FIGURE DUCT EQUIVALENTS



***Peter Franck Explains Method He's Found Useful in the Field***

*Ducts may have to be squeezed between beams and ceilings, a furred space may have to be made as flat as possible, or some unforeseen complication may arise which makes a change of sizes desirable*

THERE ARE USUALLY several places where job conditions on a ventilation or air conditioning system do not permit the installation of the ductwork exactly as laid out on the plans. Ducts may have to be squeezed between beams and ceilings, a furred space may have to be made as flat as possible, or some unforeseen complication may arise which makes a change of duct sizes desirable.

Then the problem for the engineer on the job is how to size the duct. He does not have his "equivalent rounds" chart nor his ASHVE Guide with him, so he usually figures the revised duct for the same area as the duct specified and adds a little for a high aspect ratio (ratio of duct width to duct depth). This common method appears to be the best possible, for even if he knows the formula for the equivalent round duct size,

$$d = 1.265 \sqrt[5]{\frac{(ab)^3}{a+b}}$$

he can hardly be expected to compute the $d$ of the round duct and then solve this cubic equation for the remaining side of the duct.

The purpose of this article is to provide a simple way of making the necessary computation on the job with only the help of the slide rule. The method is based on an application of the "iteration method" to the problem of finding a new rectangular duct size, say $c \times d$, when the original size, $a \times b$, is given.

By applying the above equation to both duct sizes, we find that

$$d = 1.265 \sqrt[5]{\frac{(ab)^3}{a+b}}$$

$$= 1.265 \sqrt[5]{\frac{(cd)^3}{c+d}}$$

$$\frac{(ab)^3}{a+b} = \frac{(cd)^3}{c+d}$$

$$cd = ab \sqrt[3]{\frac{c+d}{a+b}}$$

The "iteration method" is applied by making a reasonable assumption for $d$ ($c$ is assumed to be given), substituting on the right hand side of the last equation, and thus obtaining an improved value of $cd$. By repeating this procedure, the values of $d$ obtained in successive operations will quickly converge to the correct value given by the charts and tables of the ASHVE Guide.

**WHEN DUCT SIZES must be changed in the field (usually you don't have a chart or handbook with you) the task may be done quickly and easily by the method described here. All you need is the original size of the duct and a slide rule. The author —of Armo Cooling & Ventilating Co.—has found this method of computation very useful in his own field work, believes other readers will find it helpful**

It is natural to use as a "reasonable assumption" a duct size of equal area to the given duct. The result would then be obtained as follows:

*Example 1:* Given a duct 20x20 in., find the proper width of an equal friction duct with a depth of 8 in.

*Answer:* $20 \times 20 = 400$ (area of original size).

Step 1: $400 \div 8 = 50$ (this gives our "reasonable assumption").

Step 2: (evaluating root in equation):

$$\sqrt[3]{\frac{50+8}{20+20}} = 1.13.$$

Step 3: $400 \times 1.13 = 453$. (This is improved area of new duct size.)

Repeating the procedure,

$$453 \div 8 = 57.$$

$$\sqrt[3]{\frac{57+8}{20+20}} = 1.18.$$

$$400 \times 1.18 = 470.$$

Again:

$$470 \div 8 = 59.$$

$$\sqrt[3]{\frac{59+8}{20+20}} = 1.19.$$

$$400 \times 1.19 = 476.$$

Finally:

$476 \div 8 = 60.$

$$\sqrt{\frac{60+8}{20+20}} = 1.193.$$

$400 \times 1.193 = 477.$

We see by the small change of areas between 476 and 477 that no further change will be effected; the correct answer is therefore $60 \times 8$ in.

An advantage of the "iteration method" is that it is self-correcting; any mistake made in a computation will be erased by the ensuing steps. We may therefore be more daring in our initial guess. Thus, I would tackle the problem given above as follows:

*Example 2:* Same problem as in Example 1.

*Answer:*

$20 \times 20 = 400$, and $400 \div 8 = 50$.

Since this would make quite a high aspect ratio, the correct size will have to be considerably larger. Instead of $8 \times 50$, let's try $8 \times 60$:

Step 1: $60 \times 8 = 480$.

Step 2:

$$\sqrt{\frac{60+8}{20+20}} = 1.193$$

Step 3: $400 \times 1.193 = 477$.

$477 \div 8 = 59.6$

The difference between our guess and the improved answer is so small that $60 \times 8$ in. is sufficiently accurate.

In order to show how this method corrects errors in judgment, let's try one more example of the reverse problem with a reasonable (but exaggerated) initial guess:

*Example 3:* Given a duct $42 \times 5$ in., find the proper depth of a duct with equal friction having a 10 in. width.

*Answer:* $42 \times 5 = 210$, and $210 \div 10 = 21$.

Since the duct is going to be more nearly square, we guess at the size $15 \times 10$ instead of $21 \times 10$:

Step 1: $15 \times 10 = 150$.

Step 2:

$$\sqrt{\frac{15+10}{42+5}} = 0.811$$

Step 3: $210 \times 0.811 = 170$.

Since the original guess was too small, and the tendency is toward an increase, I choose $18 \times 10$ for the second improvement instead of $17 \times 10$:

$18 \times 10 = 180$

$$\sqrt{\frac{18+10}{42+5}} = 0.841.$$

$210 \times 0.841 = 177.$

$177 \div 10 = 17.7.$

Comparing 177 to 180, we see that the improvement is very small; $18 \times 10$ in. is therefore the correct answer.

# Aluminum Gages for Ducts

RECOMMENDATIONS FOR gages of 2S-¾H aluminum that can be substituted for gages of galvanized for heating, ventilating, and air conditioning ductwork of all types are given in the accompanying table. This is the alloy and temper recommended for general ductwork.

**Equivalent Galvanized and Aluminum Sheet Thicknesses for Ductwork**

| Galvanized | | Aluminum | |
|---|---|---|---|
| M.rs. Std. Gage No. | Thickness Inches | B & S Gage No. | Thickness Inches |
| 18 | 0.0478 | 13 | 0.07196 |
| | | 16 | 0.05082 |
| 20 | 0.0359 | 16 | 0.05082 |
| | | 19 | 0.03589 |
| 22 | 0.0299 | 17 | 0.04526 |
| | | 20 | 0.03196 |
| 24 | 0.0239 | 19 | 0.03589 |
| | | 22 | 0.02535 |
| 26 | 0.0179 | 22 | 0.02535 |
| | | 25 | 0.01790 |
| 28 | 0.0149 | 24 | 0.02010 |
| | | 26 | 0.01794 |
| 30 | 0.0120 | 25 | 0.01790 |
| | | 28 | 0.01264 |
| 32 | 0.0097 | 27 | 0.01420 |
| | | 30 | 0.01003 |
| 34 | 0.0082 | 28 | 0.01264 |
| | | 31 | 0.008928 |

Some 2S-H material in 0.016 in. thickness is also used for ducts, and in 0.008 in. thickness for covering joists to make cold air return ducts. Large industrial air conditioning trunk lines sometimes employ 3S-H in 0.032 and 0.040 in. thickness, with seam type corners for added strength.

Vent and duct structures mounted outside a building are usually of 0.040 in. thick aluminum to withstand wind and snow loadings.

Where a rigidity equal to that of galvanized is needed, going to the thick end of the aluminum range will take care of that requirement. However, equal rigidity is seldom necessary and often can be obtained more economically by varying the structural design of the ductwork.

For example, a large shop making ducts for commercial and industrial air conditioning installations has standardized upon a single thickness of aluminum for all its work—0.025 in. This shop finds that the economies in lower inventories, reduced handling, and the almost complete elimination of scrap more than offset the cost of going to a more rigid structural design for ducts of large section.

Use of standing seams on ducts over 24 in. wide adds rigidity. Diagonally braking large areas also stiffens the duct against "ballooning" and "oil can" effects. Longitudinal braces of "L" or "U" section bent up from sheet greatly increase rigidity, especially when tied in with standing seams.

Such designs are quite effective. For example, another shop producing large ventilating ducts reports that ducts up to 4 ft in width of ample stiffness can be made from 0.020 in. aluminum by using 24 in. wide coils and making standing seams on 21 in. centers. The standing seams are locked by dimpling with a hand tool.

Usual duct designs employ standing seam cleats across top and bottom, drive or "S" cleats on sides. The tendency of aluminum to "stick" here is alleviated by making a somewhat looser lock on the drive cleats.—G. W. BIRDSALL, Reynolds Metals Co.

**Large aluminum return duct for cooling tenant space in the Merchandise Mart, Chicago. The girl has nothing to do with the installation**

# Smoking Tendency Determines Burning Oil Performance

### *G. T. Kaufman, Technical Secretary, Oil-Heat Institute, Explains Combustion Test Unit*

BEFORE THE coming of cracking stills, oil refiners processed oil in simple distillation stills called fractionating towers, where various portions, or fractions, were separated from each other according to their boiling temperatures. Oils so separated were quite similar in their other physical properties, and their burning properties were directly tied to their boiling temperatures—the lower the boiling temperatures, the better the burning properties. These oils were generally good "burners."

With the war came the need for enormous quantities of aviation gasoline of high antiknock rating, which the fractionating towers couldn't produce. The cracking stills had to take over, particularly the catalytic cracking stills, which produce base stocks having high antiknock rating themselves, and permitting processing and blending to produce extremely high antiknock gasoline. Once in use, the catalytic crackers were found to have many advantages for the refiners and they are continuing to use them, although the demand for high octane gasoline no longer exists.

The fuel oils produced by the catalytic cracking plants are considerably different from those produced by the straight-run plants. Catalytic oils of the same boiling temperature may have great differences in gravity, for example, and in their other physical properties. Their burning qualities do not tie into any of their easily obtainable physical properties, as those of the straight-run oils do.

From time to time the trade publications have carried articles describing methods of determining burning qualities by correlating performance in actual burning with one or more physical properties. Perhaps the most important thing to observe about these proposed methods of establishing a burning quality yardstick is that the one burning characteristic common to all is smoking tendency. In all of the tests on which they are based the one thing that determined where satisfactory performance ceased was when the smoke intensity passed a certain arbitrarily established permissible maximum. This was the standard of satisfactory performance in the ASTM series of tests also. It was likewise the standard of satisfactory performance in a long series of tests on vaporizing burners conducted by the Western Petroleum Refiners Association during 1945.

**The fuel oils produced by the catalytic cracking plants are considerably different from those produced by the straight run plants; their burning properties do not tie into any of their easily obtainable physical properties**

Since smoking tendency is the universally used limiting factor in determining the satisfactory performance of burning oils, it seems logical to employ this tendency as a direct measure of any oil's degree of quality as a burning oil. In doing this we must assume that carbon forming tendency, which is of equal importance with smoke forming tendency to the vaporizing burner man, will go hand in hand with smoking tendency. No proof exists that these two will go hand in hand, but since soot is the thing that gives smoke its color and since soot is carbon, it seems reasonable to assume that they would. It would at least be more reasonable to assume that they would than that they would not; it would be hard to conceive of any oil having pronounced smoke forming tendencies not having equally pronounced carbon forming tendencies and vice versa.

When we say that oils having similar physical properties but different burning properties will burn in various types of burners with results that will be satisfactory in direct proportion to the smoke forming tendency of the oil, we say in effect that all it is necessary to do to test the burning quality of an oil is to burn it in a burner and measure and observe the results. It is proposed that the combustion reference test unit, developed by the Oil-Heat Institute, be a standard burner for this purpose.

This unit is a model, or prototype, high pressure oil burner of the simplest kind, on which all adjustments are calibrated, instruments are provided for measuring all known burning qualities, and observation windows are fitted to permit observation of what is going on. Consequently this unit can well become a universally applicable instrument for the measurement of the burning properties of the lighter grades of burning oils.

Essentially, the unit is a miniature boiler-burner unit, although the boiler is scarcely a boiler but really a cylindrical, horizontal, water cooled combustion chamber, into one end of which the burner fires, and from the opposite end of which the combustion gases travel into a smoke box, where they may be observed by means of light provided and their smoke content measured by smoke meter. In the end of the combustion chamber opposite the burner a water cooled steel vessel is placed. This may be moved toward and away from the burner by a handwheel attached to a rack and pinion, and supplies means for regulating the length of the combustion chamber. In the sides of the cylinder are numerous observation ports, sealed with pyrex glass, which permit direct observation of flame length, flame color, and so on.—*Excerpts from a paper given at the annual meeting of the Oil-Heat Institute.*

By F. W. HUTCHINSON

**A SIMPLE, straightforward method of extending the pressure-enthalpy chart concept of ideal refrigerating cycle representation to include those deviations which most frequently occur in actual cycles is given here. This chart is particularly suitable for use in showing such deviations since, without exception, they can all be expressed in terms of equivalent series of constant enthalpy and constant pressure processes and hence appear on the chart as either horizontal or vertical lines. To demonstrate this fact more forcefully, an exaggerated example illustrative of actual cycle deviations is presented in detail. The author is professor of mechanical engineering at Purdue University, and this technical paper was prepared in cooperation with the housing research division of the Purdue Research Foundation—G. Stanley Meikle, research director; Carl F. Boester, housing executive.**

# *Analyzing* REFRIGERATION CYCLES

ONE OF THE most effective aids available to the engineer for the thermodynamic analysis of refrigeration cycles is the pressure-enthalpy plotting of refrigerant properties.* Where the steam power engineer prefers the temperature-entropy plotting and the internal combustion man favors pressure-volume diagrams, the refrigeration engineer almost inevitably selects pressure-enthalpy as the form best adapted to his needs.

On such a chart, unit area does not have any practical significance, but for the ideal reversible cycle it is possible to visualize both heat (gain in evaporator and loss in condenser) and work as proportional to linear distances on the chart. Further, since both evaporator and condenser processes occur at constant pressure and the expansion valve process occurs at constant enthalpy, it follows that complete representation of the ideal cycle can be accomplished on the pressure-enthalpy chart by means of vertical and horizontal lines—excluding only the compression process which, for the ideal case, is an isentropic. A detailed discussion of theoretical cycle analysis by means of the pressure-enthalpy chart is given in chapter 2, Compression Systems, of the 1942 edition of the American Society of Refrigerating Engineers Data Book.

The present article is intended primarily as a supplement to the ASRE Data Book in showing how readily the *ph* (pressure-enthalpy) chart is adaptable for use in representing the multitude of deviations from ideal conditions which inevitably occur in any actual system. As will appear in subsequent discussion, all deviations from the theoretical reversible cycle except those which occur during the actual process of compression can be represented by either vertical or horizontal lines. Thus, the *ph* chart affords the simplest possible means of showing exactly the thermodynamic changes which occur during even the most complex of actual operating cycles.

### Types of Deviations from Ideal Cycle

The fundamental assumptions underlying the ideal cycle are that no pressure change occurs other than in the compressor or the expansion valve and that no gain or loss of heat takes place except during passage of the refrigerant through the condenser and evaporator. A further assumption is that the compression process is isentropic, hence occurs without energy transfer to or from the working fluid other than that introduced into the refrigerant as the heat equivalent of the shaft work. In any actual cycle there are, of course, friction losses in the piping and in the individual pieces of equipment which are responsible for a progressive drop in pressure of the refrigerant from the point of discharge from the compressor

*Definitions of some of the terms used in this article:

Adiabatic—A process of compression which takes place without transfer of heat to or from the vapor within the cylinder. No actual compressor is adiabatic but most modern high speed machines perform very nearly adiabatically if the cylinders and cylinder heads are not cooled.

Enthalpy—A thermodynamic property which is equal to the sum of the internal energy (so-called "heat" energy) of the vapor plus the amount of flow work required to maintain steady and continuous fluid motion.

Entropy—A mathematical property used to determine changes in the degree of availability of energy associated with a working substance. In refrigeration, the absolute value of entropy is unimportant but an entropy increase indicates loss of availability while an entropy decrease indicates increase of availability. The entropy of a fluid may decrease but only at the expense of an entropy increase for the system as a whole.

Isentropic—An adiabatic process which occurs without change of entropy. Such a process is subject to simple mathematical description and is a conservative first approximation to actual refrigeration compressor performance.

Pressure—Usually thought of as a force per unit area (as pounds per square foot) but the thermodynamic significance of pressure is associated with the fact that it has the units of foot-pounds per cubic foot, hence is numerically equal to the flow work associated with one cubic foot of steadily moving fluid.

Reversible—A process which occurs without change of entropy; a process which occurs without friction or other loss.

Wiredrawing—The pressure loss associated with passage of a fluid through a valve.

through the system to the suction entrance of the compressor. Further, the connecting piping inevitably passes through spaces which are at greater or lesser temperatures and which are, therefore, responsible for heat transfer to or from the refrigerant.

The actual compression process is likely to be markedly different from the ideal. During constant pressure admission, the low temperature suction gases are first subject to wiredrawing as they pass through the suction valves of the compressor and then undergo an increase in internal energy as heat flows from the higher temperature cylinder walls to the cold gases. During this period, the internal energy, enthalpy, and entropy of the superheated refrigerant vapor is increasing. During the first part of the actual process of compression, the gases are at a temperature less than that of the cylinder walls; hence, energy continues to flow as heat from walls to refrigerant.

As the compression process continues, the gas temperature increases (due to the transformation of shaft work to internal energy) until eventually the gas within the cylinder is at the same temperature as the cylinder wall. At this particular instant, the rate of energy transfer from wall to gas becomes zero and from here on the flow of heat is in the opposite direction. As the pressure increases, the temperature difference between gas and wall likewise increases until at the time the discharge valves open, the rate of energy transfer from gas to wall is a maximum. Throughout the subsequent period of constant pressure discharge, the gas continues to lose heat to the relatively cold cylinder walls; at the same time, wiredrawing through the discharge valves serves to reduce the pressure at constant enthalpy so that the actual refrigerant at a point in the line outside the compressor has both a lower pressure and a higher enthalpy than it had within the cylinder at the time the discharge valves opened.

If a compressor cylinder were perfectly insulated, there would be no net gain or loss of energy to the surroundings; in practice, however, the cylinder is not merely imperfectly insulated but is in many cases provided with some special means (as a water jacket) for assuring a net transfer of energy as heat from the cylinder wall to the surround. In such cases, and irrespective of the energy added to the refrigerant as a result of shaft work, the heat received by the gas from the cylinder during admission and during the early part of the compression process must inevitably be substantially less than that delivered by the gas to the cylinder during admission and during the latter part of compression and during discharge.

At first glance, it would appear that this net heat loss would effectively prevent application to the actual compressor of the principle of reversibility, yet it can be readily shown that only when there is a net loss can the unit be expected to perform with reasonable fidelity

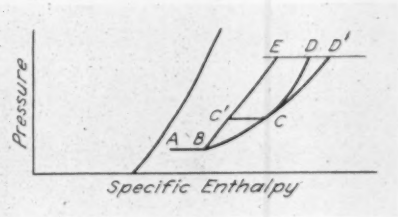



**Fig. 1—Energy transfers which occur during a compression process in which there is no net energy transfer from the compressor**

to the reversible pattern. Paradoxical though this may seem, it occurs because of the fact that in a perfectly insulated compressor cylinder, there would necessarily be an increase in entropy of the refrigerant due to the fact that the entropy gain associated with each Btu transferred from wall to gas would be greater than the corresponding entropy loss experienced when a Btu is transferred back (at higher temperature) from the gas to the cylinder wall. Thus, the effect of water jacketing is to tend to restore the entropy of the refrigerant at discharge from the compressor to the value which it would have had if the compression process (including admission and discharge) had been isentropic.

Fig. 1 illustrates the energy transfers which occur during a compression process in which there is no net energy transfer from the compressor. During the constant pressure admission, the enthalpy increases from $h_A$ to $h_B$. Subsequently, during the first part of the compression process, energy is received at a constantly decreasing rate until, at point $C$, the gas is at the same temperature as the cylinder walls and has received an enthalpy increase during the first part of compression amounting to $h_C - h_B$, of which $h_C - h_{C'}$, is a gain in excess of that which would have occurred if the process had followed the isentropic line $BC'$.

During the latter part of the process, the compression would follow $CD'$ if it were isentropic but due to the continuous and increasing heat loss to the cylinder wall, the actual enthalpy gain of the gas is reduced from $h_{D'} - h_C$, the isentropic value, to $h_D - h_C$. The added energy gain during the early part of compression, $h_C - h_{C'}$ may be exactly equal to the reduction in gain during the latter part, $h_{D'} - h_D$, but even so the entropy at point $D$ will exceed that at point $B$ because of the fact that the isentropic $CD'$ is of flatter slope than the isentropic $BC'$. If water jacketing were provided, sufficient heat could be removed to reduce the enthalpy of the discharged gas from $h_D$ to $h_{D'}$, thereby restoring it to the state point which would have existed at discharge if the compression had been reversible and no energy transfer had occurred between the gas and the cylinder wall.

Note, however, that even though the state point has been restored to the isentropic discharge value, it does not follow that the work of compression has been reduced to its isentropic magnitude; by the General Energy Equation, $(h_E - h_B) + Q_{jacket} = Wk_{compression}$ where $Q_{jacket}$ is the heat loss to the jacket expressed in Btu per lb per min of refrigerant passing through the compressor and $Wk_{compression}$ is the energy input to the compressor (neglecting mechanical losses) expressed in Btu per lb of refrigerant per min; $(Wk_{compression} \div 42.42)\ W$, where $W$ is the refrigerant flow rate in pounds per minute, is the horsepower required for the system.

### The Actual Cycle Diagrammed

Fig. 2a shows an equipment arrangement corresponding to an actual system while Fig. 2b shows

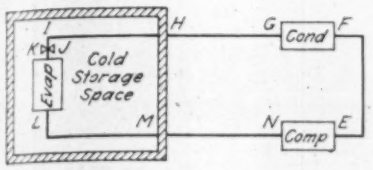



**Fig. 2a—Equipment arrangement corresponding to an actual system. Fig. 2b shows the pressure-enthalpy representation of the cycle**

the *ph* representation of the cycle. Starting with the refrigerant at state *A* just after passage through the suction valves of the compressor, the state changes during admission, compression, and discharge are shown by the path *ABCD*, all as described in detail in the preceding section. From state *D* the refrigerant passes through the discharge valves according to the throttling process *DE* and its state in the line just outside the compressor is then *E*. From here the gas passes through the line to the condenser and probably undergoes some desuperheating as well as a slight pressure drop due to friction; for purposes of *ph* representation these two losses, of heat and of pressure, can be considered separately, thus permitting representation of the actual complex process as consisting first of the desuperheating constant pressure process *EE'* followed by the equivalent throttling process *E'F*. Regardless of which of the two equivalent processes is considered as occurring first, the end state of the refrigerant—at entrance to the condenser—must, of course, be the same: *F*.

During passage through the condenser, the refrigerant continues to lose heat until it reaches a saturated vapor state, then undergoes condensation, and finally may be somewhat subcooled; these three heat dissipation steps can all be grouped and represented by the single equivalent constant pressure process *FF'*. The pressure loss which occurs during passage through the condenser is usually too small to be of practical significance, but, if desired, it can be shown as an equivalent constant enthalpy process *F'G*. Between condenser discharge and entrance to the cold storage room, the subcooled liquid may pass through spaces at higher temperature and from which heat will be received. If this is the case, the line process will be shown as an equivalent heat gain, *GG'* followed by equivalent line pressure drop, *G'H;* note that re-vaporization of the refrigerant, with consequent loss of refrigerating effect and reduction in the coefficient of performance, may then occur.

On entrance to the cold storage room, the refrigerant passes through the length of pipe *HI* which is located ahead of the expansion valve. Since the room is at lower temperature than the refrigerant, there will be some cooling effect, *HH'* and the usual line pressure loss, *H'I*. Note that even if the enthalpy loss from *H* to *H'* were exactly equal to the enthalpy gain from *G* to *G'*, there would be no regain of efficiency since such cooling as occurs along the line

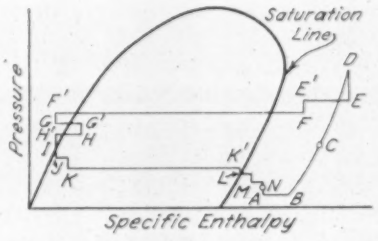



**Fig. 2b—Pressure-enthalpy (*ph*) representation of refrigerating cycle diagrammed in Fig. 2a**

*HI* is at the expense of increased evaporator load. However, if this energy quantity is added to evaporator load in figuring the capacity of the system, it should be subtracted when figuring condenser capacity since the added load does not appear at the condenser. In effect, this energy quantity serves only to increase the required size of the evaporator.

From *I* to *J* the fluid passes through the expansion valve. From *J* to *K* it picks up heat from the cold storage space and loses pressure due to line friction. The evaporator process is shown by the steps *KK'* and *K'L*, the refrigerant being assumed a saturated vapor at discharge from the evaporator. In the piping between evaporator and the wall of the cold storage room, additional useful heat is picked up, *LM*, while between the cold storage room and the compressor such heat gain as does occur, *MN*, represents an added compressor and condenser load for which there is no compensating increase in refrigerating effect. For this system, as shown, the actual energy gain of the refrigerant as it passes through the cold storage space is $h_M - h_I$, but the effective refrigerating effect is $h_M - h_H$.

From state *N*, in the pipe just ahead of the suction valves of the compressor, the refrigerant goes through the valves with a constant enthalpy wiredrawing loss as shown by *NA*; from this state point, the cycle is then repeated.

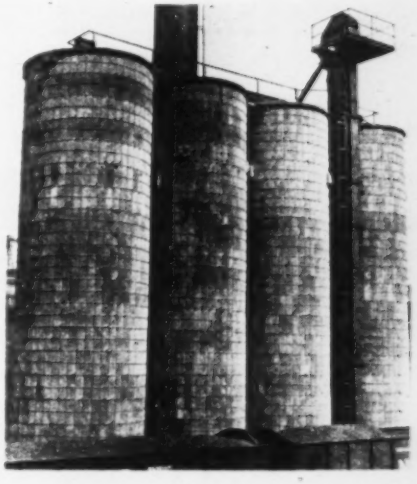
## SILOS STORE COAL FOR PLANT HEATING

More than one-fourth of all the coal used annually for heating the Superior Engine Div. plant of the National Supply Co., Springfield, Ohio, can be stored in four new concrete silos. It is the company's answer to a coal strike or any circumstances that would delay coal shipments except for an extended period.

The new silos will store 1000 tons of coal, are part of the $1,525,000 plant improvement program for Superior Engine, manufacturer of diesels and gas engines. Handling of the coal now is completely automatic from the time it is dumped from the car until it is in the furnace.

A similar system of silos and automatic handling is planned by the company for core sand storage and distribution in the plant's foundry rehabilitation program scheduled for completion late this year.

**These four new silos at Superior Engine are the company's answer to a coal strike or any circumstance that would delay coal shipments except for an extended period**



# PLANNING TO AIR CONDITION?

REGARDLESS OF THE extent of air conditioning contemplated, either for the present or the future, it should be *planned*, so that—

1) The air conditioning results required will be obtained.
2) Electric service in transformer and switching capacity will be available.
3) Electric wiring of ample capacity and adequate distribution will be available at the points required.
4) Water and drain pipes of ample capacity will be accessible.
5) Maintenance of equipment will be in accordance with a definite, prescribed plan.
6) The entire development of air conditioning within the building will be systematically planned, regardless of the type of equipment or system used, and regardless of who pays for the equipment.

It is recommended that a competent consulting engineer be engaged to investigate the designs best suited to the particular requirements and present cost estimates of the proposed methods of air conditioning, so that regardless of the extent to which air conditioning may develop or be required in the building, it will be in accordance with a carefully conceived plan.

The principal reason for air conditioning of office buildings is that of maintaining a definite standard of comfort for those occupying the building. The American Society of Heating and Ventilating Engineers defines comfort air conditioning to be "The process by which simultaneously the temperature, moisture content, movement and quality of air in enclosed spaces intended for human occupancy may be maintained within required limits."

### Performance

The first prerequisite in considering air conditioning of an office building is to determine definitely the standards that must be maintained in order that it be acceptable to the occupants of the building, by giving them the degree of comfort now obtainable with a properly designed system. The second prerequisite is to analyze carefully any or every proposed system of air conditioning for the purpose of determining definitely if it will *produce* and *maintain* those predetermined standards under *definitely specified internal conditions* regardless of the temperature or quality of the outside air.

Air conditioning for human comfort is still being refined. Present day minimum acceptable conditions may be wholly inadequate five years hence. The purchase of equipment that is known to be incapable of producing high grade results, in the light of our present knowledge of satisfactory performance, will most certainly prove to be an uneconomical investment.

The ASHVE also gives a more detailed definition of air conditioning, which provides a clue as to what may be expected in the future. This defines air conditioning to be "The simultaneous control of all or at least the first three of those factors affecting both the physical and chemical conditions of the atmosphere within any structure. These factors include temperature, humidity, motion, distribution, dust, bacteria, odors and toxic gases, most of which affect in a greater or lesser degree human health or comfort."

### Temperatures and Humidity

Regardless of the outside weather conditions, the air conditioning equipment should be able to produce a comfortable atmosphere within the building. It is desirable, for best results, to have means of controlling independently the wet and dry bulb temperatures. There will be days during which relatively little lowering of the dry bulb temperature will be required but the humidity will be abnormally high. If refrigeration is used to extract this moisture it may be necessary to reheat the air. If dehydration is used, the air may have to be cooled.

Localized control of the dry bulb temperature conditions within each office may not be necessary. Yet out of every 100 persons subjected to a definite temperature condition, some will be too hot, some too cold. Tests made by the research laboratory of the ASHVE show that for a large group of people, the feeling of comfort extended over quite a large range. In large offices, this same difference of opinion doubtless will continue to exist, but in individual offices it is possible to have equipment which will permit localized temperature adjustment in accordance with the desires of someone within that office.

The installation of equipment which will provide means for localized control of temperature may not be necessary for office buildings, but in the home, apartment house or hotel, it would definitely be a necessity. Should the trend in office buildings indicate that some time in the future it may be desirable to provide localized temperature control, then it might be well

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This Report on Planned Air Conditioning for Office Buildings has been prepared by the subcommittee on air conditioning of the committee on education and research of the Building Owners' and Managers' Association of Philadelphia. The committee presented a report on January 15 1945 which concluded with the statement that "the committee will continue the study of air conditioning and make a further report." Since that time the committee has been pooling ideas and information for the purpose of preparing a report which would create an incentive on the part of each building manager to begin immediately a serious consideration of the manner in which air conditioning is to be developed in his particular building.

The committee is indebted to and acknowledges the assistance and cooperation of the publicity committee of the air conditioning and industrial refrigeration division of the Electrical Association of Philadelphia, who, after reviewing and offering many suggestions which were incorporated in the report, have approved it; also to the Philadelphia Electric Co. for the great amount of time and thought given in conferences with the committee and for their invaluable suggestions.

The report was presented by C. Robert Cheyney, operating manager of the Fidelity-Philadelphia Trust building, at the annual meeting of the Middle Atlantic Conference of Building Owners and Managers. The first part is published here, and the rest of the report will appear in later issues.

to determine the cost of installing this additional equipment and as to whether or not it may be added to the present equipment at some future date.

### Air Motion and Distribution

Air motion and distribution are two distinct subjects, and must be treated separately, yet each depends upon the other for satisfactory results.

Properly designed air conditioning of a given space must insure the supplying to that space of a definite amount of air at definite temperatures and humidity. Calculations presuppose that the conditioned air enters in such a manner as to be distributed uniformly over the plane of entrance, and that it moves uniformly either across the room or from the ceiling to the floor.

According to effective temperature charts, the sensation of warmth or cold is induced by the combination of three conditions of the air; i.e., dry bulb temperature, relative humidity, and velocity. The higher the temperature and humidity of the air, the higher air velocity one can withstand with the same relative sensation of comfort. But at 80 deg room temperature and lower, with 50 per cent relative humidity, any perceptible air movement at the occupancy level may create a condition of annoyance or cold draft objectionable to most persons.

The quantity and quality of conditioned air liberated into a given space theoretically may be sufficient to provide the required comfort cooling, but if this air is not uniformly distributed in such a manner that all portions of the space are subjected to approximately the same velocity, some parts of the room will not be receiving the designed benefits of air conditioning, while other parts will be drafty.

### Air Entrained Impurities

The control of air quality or contaminants is one of the functions of complete air conditioning. The means to be employed in the purification of air will depend upon the nature and concentration of the entrained undesirable substances and the extent of their elimination required.

In considering the design and installation of an extensive air conditioning system, it is well to have a knowledge of those things we now recognize as air contaminants. This is essential in arriving at definite conclusions as to the provisions that must be made to accomplish prescribed results, also for the purpose of attempting to forecast the requirements five years hence. The extent to which proposed equipment meets present predetermined requirements, and the extent to which the equipment lends itself to the inclusion of devices which may be prerequisite in the near future should be given very careful consideration.

Air cleaning is the process of capturing and removing solid matter from a stream of air. Air purification deals with the removal of harmful or unpleasant gases, vapors or bacteria from the air stream or from an occupied space. Dusts, fumes, and smokes are known as solid particulate air contaminants; mists and fogs are known as liquid particulate air contaminants; vapors and gases as non-particulate air contaminants. General definitions and means for removal of contaminants are briefly discussed herewith.

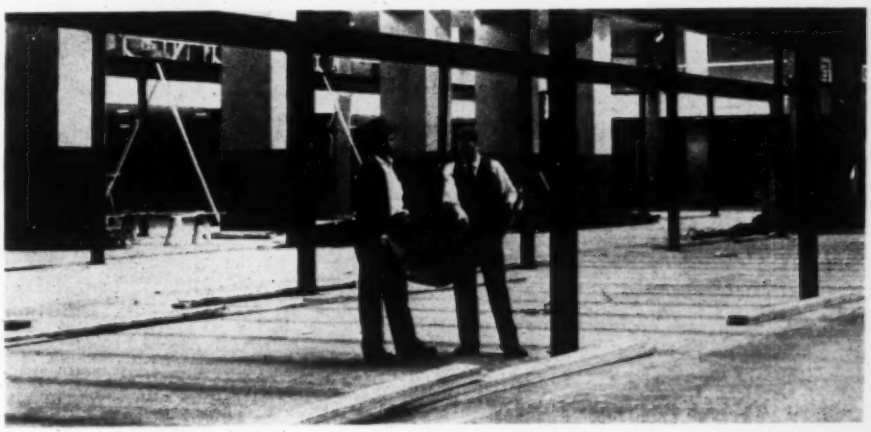
*Dust*—All solid particles of matter which are light enough to be blown about by air currents yet have enough mass or weight to settle readily by gravity in air which is comparatively still. Small particles are generally invisible to the naked eye. Molecular in size, these particles are measured in terms of microns, a metric unit 0.001 millimeter or approximately 1/25,000 of an inch. The smallest particle of dust the unaided eye can distinguish under ideal conditions is about 10 microns. In general, the lower limit of visibility is from 10 to 50 microns, depending upon the individual eye, the light intensity, and the color and shape of the particle. Street dust normally is invisible, and will average 0.5 micron, but the individual sizes may vary from 50 microns down to less than 0.1 micron.

It is the small particles of dust that are most objectionable and cause the greatest amount of destruction, in settling on walls and furniture. A filter may be capable of removing 90 per cent of the total *weight* of dust from the air flowing through it, yet only remove the larger chunks, so to speak. The remaining 10 per cent may consist of dust particles 1 micron and less in size, and are of the kind that we see deposited on the walls above radiators, around grilles, and above electric light fixtures.

From this standpoint it is obvious that the efficiency of a filter in terms of the per cent by weight of dust removed from the air stream is meaningless. The National Bureau of Standards has developed the dust-spot or blackness test for air filter efficiency. This test consists of drawing samples of cleaned air and of uncleaned air through filter papers simultaneously. By measuring the density of the blackness spots in a given length of time and the volume of air flowing, and also by measuring the volume of filtered air flowing required to produce a spot of equal density to that of the unfiltered air, the efficiency of the filter is obtained.

The extent to which dust parti-

**A competent consulting engineer should be engaged**



cles are to be filtered from the air will determine the kind of filter used. The smaller the dust particle to be removed, the more expensive the equipment. However, it may prove that the removal of the smaller dust particles will reduce cleaning and redecorating costs to such an extent that more expensive equipment will be the cheapest in the long run. Filters are available for removing large chunks of dust. Others are available for removing dust particles as small as 0.1 micron in size. What kind of a filtering job is wanted? What is anticipated as the requirement five years hence and will it be possible to utilize the required equipment effectively?

The dust problem should be given most careful consideration. The daily cleaning of offices and the periodic redecorating constitute one of the major expenses. Mechanical filters of proper design may accomplish satisfactory results. A combination of mechanical filters and water spray will eliminate still more dust. The ultimate, as of today, is the electrostatic dust precipitator.

*Smoke*—Smoke is an air suspension of particles, usually but not necessarily solid, resulting from incomplete combustion. The removal of tobacco smoke from a room may be accomplished by diluting the smoke with a sufficient amount of fresh air. So long as the amount of fresh air supply does not exceed that required by the normal occupancy of the room, no particular difficulty is experienced, but if the smoke is heavy and additional fresh air is required to maintain the required threshold condition, a point may soon be reached where some more economical means for removing the smoke may be considered. Tobacco smoke is made up of very small solid particles (0.1 to 0.3 micron) which constitute the opaque haze, and a gas which contains the odor. The solid particles can be removed by an electrostatic dust precipitator and the odor by activated carbon. Equipment is available for accomplishing both of these results, either locally or in connection with a central plant air conditioning system.

*Bacteria*—The elimination of germs and bacteria detrimental to health has been steadily pursued by science for many years. There are many ways in which air may become polluted. Human beings are the chief contaminators. By coughing, sneezing or merely talking, people expel minute droplets and particles from their noses and throats. The larger particles or droplets settle to the floor, but the smaller ones remain suspended in the air for long periods. Some bacteria are caught on floating dust particles or rise from the dust on the floor.

The control of bacteria in the air within the air conditioned space is being done quite generally by the pharmaceutical industry by means of bactericidal lamps. As to what extent this may be deemed advisable or even necessary in the future in connection with air conditioning office buildings appears to be purely speculative at this time, unless one is gifted with inspired foretelling of future developments and requirements. However, it is always advisable to attempt to anticipate future developments and then decide upon the advisability of making provision for adding the required additional equipment to the present equipment at the least expense.

In attempting to foretell future trends, we might consider, for example, the possible development in air conditioning toward reducing the spread of communicable diseases. At the present time, the purity of the outside air, from a bacteriological standpoint is, in general, considered satisfactory so far as the safety of the human being is concerned. Conceivably, this same air, when delivered to an office building, distributed to the various offices or groups of offices, intermixed, and again recirculated may pick up bacteria from one office and convey them to some other office. This condition may be prevented through the use of a system which recirculates the air within only one office, or through the use of bactericidal lamps installed in the air stream at one or more central points. Electrostatic dust precipitators will remove bacteria attached to dust particles.

# EXPECT MORE PIPE FITTING PRODUCTION

MORE THAN a 10 per cent increase in output of critically scarce pipe fittings will mark 1947 if production schedules are not interrupted in the 36 plants in the industry, according to R. L. Stewart, president of the Pipe Fittings Manufacturers Association. Factory stocks and jobber stocks are bare, Mr. Stewart said, and industry-wide inventories do not exist, because pipe fittings are shipped as rapidly as they are produced.

Six major problems facing the industry are listed by Mr. Stewart as shortages of pig iron, inequitable allocations of pig iron in southern as compared with northern regions of production, pressure for adoption of international pipe thread specifications from British sources, supplying products to new distributors, meeting greatly expanded demand for fittings from industrial buyers, and freight car shortages. Despite these matters, an improved outlook has been indicated by a noticeable increase in man-hour productivity of workers, he declared.

American pipe fitting manufacturers are strongly opposed to accepting the proposal of British manufacturers to adopt an international pipe thread specification for standardization, unless American pipe thread standards are used as the basis of agreement, according to the association. Spokesmen said that it would cost American pipe fitting manufacturers more to change over now than all the remaining plants in the world.

# Unit Heaters Meet Architectural Requirements of Modern Store

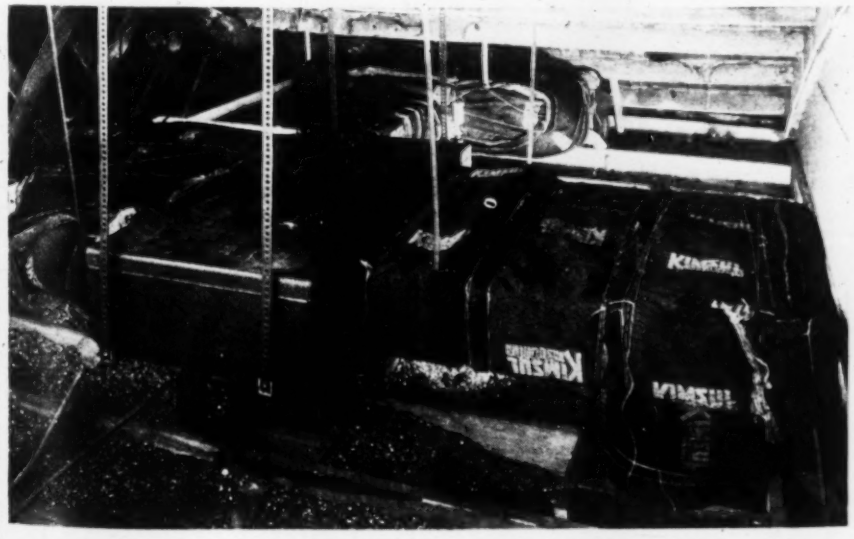
*TWO CABINET unit heaters, installed in attic space on either side of the entrance to a drapery store, supply air through ceiling outlets. The scheme keeps the show windows frost-free and permits full use of valuable display space. Robert V. Williams, of Modine Mfg. Co., describes the setup. This attractive store is a modernized commercial garage. The two ceiling air outlets may be seen on either side of the door.*

FULL LENGTH WINDOWS—a popular trend in today's commercial architecture—afford the retail store an excellent opportunity to construct attractive merchandise displays. Unless the display space is free of distracting heating equipment, however, its full value will not be realized.

A difficult problem presented by this type of window is that of heating the areas adjacent to it and keeping the window devoid of frost in cold weather. Christensen's Draperies, Inc., Racine, Wis., solved the problem by installing two cabinet unit heaters above the false ceiling and connecting the units by ducts to venturi openings in the ceiling. These units are located above bays at either side of the entrance door. Although together they have a capacity of 900 sq ft of radiation, they occupy only a little over 18 cu ft of space.

One of the pictures shows one of these units. It is suspended from the roof construction, as is the ductwork. The round duct leading downward at the right hand side of the picture leads to the venturi opening in the ceiling of the store proper. This is fed by a duct connected directly to the unit's outlet grille. The duct visible in the upper center of the photograph and to the left of the unit itself carries the returning cooler air from a grille opening near the floor in the store proper. This cooler air is fed directly to the intake grille of the unit. For photographic purposes, the insulation was removed from the cabinet unit heater so as to make it more distinguishable from the surrounding ductwork. A setup exactly like this one is installed on the other side of the attic room.

**One of the two unit heaters and connecting ductwork in the attic space. The insulation was removed from the unit for this photograph**



An appearance of spaciousness in the store is made possible through the lights which are mounted flush against the ceiling. The pattern of lights and the effect they produce is aided by the circular venturi outlets painted to blend with the color scheme.

The balance of the store is heated by convectors, and both the cabinet unit heaters and the convectors are installed on a forced hot water system.

Architectural work was done by Al Seitz and the heating equipment was installed by Wiertz & Hughes.

# Fluid Flow's Relation to Installation & Operation

## *Undersize Refrigerant Piping a False Economy, Says H. E. Ferrill*

**NEGLECT OF ENGINEERING fundamentals may penalize otherwise good equipment with a faulty installation, causing headaches for all concerned. Mr. Ferrill, who is chief engineer of the brass division of the Kerotest Mfg. Co., shows here that undersized refrigerant suction piping is a false economy, for it can save at the most about 3 per cent of the complete installation but in so doing will increase operating and maintenance costs from 6 to 38 per cent**

AN ERROR IN THE size of refrigerant piping can impose severe penalties on the capacity of the condensing unit—sometimes as much as 30 per cent. Such an error is not immediately apparent and is extremely difficult to determine on the average installation, and there is no convenient adjustment to correct it or compensate for it. The importance of proper sizing of the piping in the original layout is thus readily seen.

### Analyzing Typical Installation

In order to set up and analyze a typical installation, certain assumptions are necessary. Those listed below, and the tabulated capacities, are typical of modern air conditioning equipment and do not represent the equipment of any particular manufacturer or single installation.

Total Btu load: 200,000 Btu per hr.
Refrigerant: "Freon 12."
Refrigerant temperature in evaporator: 35 F.
Motor horsepower: 20.
Compressor: 4 cylinder, 4 x 4, 500 rpm.
Heat exchanger pressure drop: 5.0 psi.
Suction line tubing: 1⅝ in. OD.
Equivalent length of suction line: 130 ft.
Line pressure drop (see chart, page 80, March HPAC): $130/100 \times 3.2 = 4.2$ psi.
Total pressure drop: 9.2 psi.
Compressor inlet pressure: 23.37 psig.

From the above assumptions it is possible to compute a curve representative of the relation between the condensing unit capacity and suction pressure. This requires the additional assumptions of a constant condenser temperature and a constant volumetric efficiency, neither of which is actually the case. However, the errors introduced by these inaccuracies are minor and, in any event, the computed values shown in the graph may be considered typical.

Computations are on the basis of rpm, piston displacement, and the specific volume and latent heat content of "Freon 12" refrigerant. It is interesting to note that the compressor moves the same *volume* of vapor throughout the entire range of plotted capacity. The capacity is reduced at lower pressures since the density of the vapor is lower and there is a consequent reduction in the number of pounds of refrigerant condensed.

Referring to the assumed compressor specifications, we can compute the piston displacement and the volume of refrigerant moved as follows:

$$V_{hr} = \frac{\text{Rpm} \times D \times E_v \times 60}{1728}$$

where

$V_{hr}$ = volume of vapor moved, cu ft per hr;
Rpm = speed (500 rpm assumed);
$D$ = total piston displacement, cu in. per revolution, $= [(4 \times 4 \times 4 \times 4\ \pi)/4]$;
$E_v$ = volumetric efficiency, 90 per cent (assumed).

From the above, $V_{hr} = 3140$ cu ft per hr.

The capacity of the condensing unit may then be computed from the latent heat content of the refrigerant vapor in Btu per cu ft at various pressures:

Capacity (Btu per hr) $= V_{hr} \times h/v$ where

$h$ = latent heat of vaporization;
$v$ = specific volume.
$h$ and $v$ vary with pressure and temperature and are obtained from thermodynamic tables.

These values have been computed and the results represented graphically herewith. It should be noted that the capacity varies almost directly with the suction pressure, the slight deviation being caused by the variation of latent heat content with pressure.

The operating condition, then, of the condensing unit may be represented by point $A$; that is, 200,000 Btu per hr capacity at a suction pressure of 23.37 psig. Continuing with the assumptions previously noted, the horizontal distance between $A$ and $B$ may be taken to represent the total line and heat exchanger pressure drop (9.2 psi) and the point $B$ the operating condition of the evaporator; that is, 200,000 Btu per hr at a pressure of 32.57 psig.

### Effects of Suction Line Pressure Drop

A complete investigation of the effects of this suction line pressure drop can now be made. Since the resistance to flow encountered must be overcome in order to move the gas to the compressor, some energy must be expended and the only source of this energy is the compressor motor. Some percentage of the total motor horsepower is used to maintain the velocity of the suction vapor in the suction line.

This power expenditure may be computed by an analysis of the heat content of the suction vapor at the two ends of the suction line (excluding the heat exchanger). If the heat exchanger is reasonably efficient the suction vapor will be superheated to approximately 65 F by the time it enters the suction line

proper. Since the temperature difference between the suction vapor and the surrounding air is small and the time involved is extremely short, we may consider the expansion through the suction line to be at constant temperature. It will not be adiabatic since some heat is added by the friction against the walls of the tubing. The condition of the vapor at the two ends of the line will be:

*Heat exchanger end*
$P = 27.57$ psig
$T = 65$ F
$H = 86.76$ Btu per lb
*Compressor end*
$P = 23.37$ psig
$T = 65$ F
$H = 86.93$ Btu per lb

In order for the system to have the capacity of 200,000 Btu per hr, a total of 3020 lb of refrigerant must be moved (Btu load/latent heat per lb). The total heat gain is then $3020 \times (86.93 - 86.76) = 514.0$ Btu per hr.

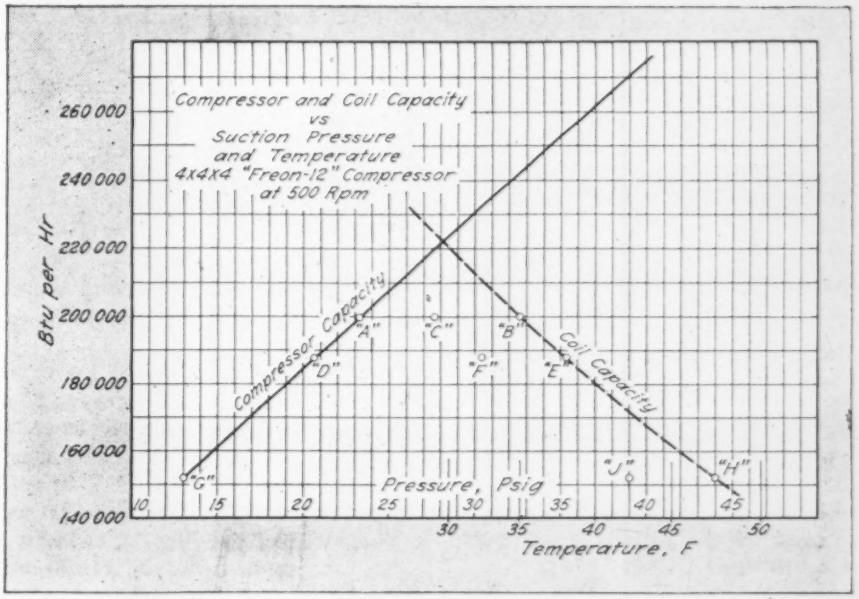
Converting to mechanical and electrical power, $514 \times 0.000393 = 0.202$ hp, or $514 \times 0.000293 = 0.1505$ kw. This expenditure of power represents approximately 1 per cent of the total motor horsepower. This is not an unreasonable price to pay for proper lubrication of remote parts of the system, brought about by good oil circulation.

The foregoing analysis is based on good practice, proper attention having been given to the effects of suction line resistance. It should be noted that the suction line pressure drop in this case is 3.2 psi per 100 ft. A 1⅝ in. OD suction line was chosen in preference to a 2⅛ in. OD line, since the selection of the larger would have resulted in a rather low velocity.

### Smaller Size Suction Line

If, for fancied economy or for any of a number of other reasons, the suction line had been reduced in size, the cost of overcoming the increased resistance would have increased rapidly. Assuming that a 1⅜ in. OD line were installed, a similar analysis may be made.

From the chart on page 80 of the March HPAC. we determine the line pressure drop in this case to be 7.5 psi per 100 ft, or $7.5 \times (130/100) = 9.75$ psi total. Adding to this the heat exchanger drop of 5.0 psi, we have a differential of




14.75 psi existing between the coil outlet and the compressor suction.

Referring now to the accompanying graph, we may represent the first analysis by the horizontal line *ACB*, point *A* representing the condition at the compressor suction; point *C* that at the heat exchanger outlet; and point *B* that at the evaporator outlet. Since the capacity of the evaporator is almost directly proportional to the temperature difference between the refrigerant and the air, we may represent the coil capacity by a curve similar to the dotted line. If this curve represents evaporator capacity, we may then determine the operating conditions enforced by the use of the smaller suction line.

The total pressure drop in this case has been determined as 14.75 psi. Since the evaporator and the condensing unit were balanced before, it is obvious that the operating pressures of each will be different under these new conditions. These conditions may be represented by the line *DFE*, the pressures at *D* and *E* differing by 14.75 psi, the total pressure drop, and those at *F* and *E* by 5.0 psi, the pressure drop due to the heat exchanger.

That the system will balance at these conditions is readily apparent. Since the pressure drop in the suction line is increased, the compressor will operate at a lower suction pressure with correspondingly reduced capacity. Since the evaporator cannot absorb more heat into the system than the condensing unit can remove, it will of necessity operate at a higher refrigerant temperature, where its capacity exactly matches that of the condensing unit.

The condition of the suction vapor at the two ends of the suction line, then, will be:

*Heat exchanger end*
$P = 30.4$ psig
$T = 65$ F
$H = 86.64$ Btu per lb
*Compressor end*
$P = 20.6$ psig
$T = 65$ F
$H = 87.24$ Btu per lb

This heat gain of 0.60 Btu per lb equals a total of $3020 \times 0.60 = 1812$ Btu per hr.

Converting again to power, $1812 \times 0.000393 = 0.72$ hp, or $1812 \times 0.000293 = 0.531$ kw. This new power expenditure represents 3.6 per cent of the total motor horsepower and it is interesting to note that it has more than tripled over the previous instance.

### Condensing Unit Capacity Reduced

This is not, however, the most serious effect of the small suction line. Referring again to the graph, we find that our condensing unit capacity is reduced to 188,000 Btu per hr by the reduced suction pressure. This represents a loss of 12,000 Btu per hr, or 6 per cent of the original capacity. Hence, if the system has been designed to operate 100 per cent of the time under full load conditions, it will now be 6 per cent undersize. Accordingly, to handle the same load in a 24 hr day under varying load conditions it

will be required to operate 6.4 per cent more of the time. Considering the increased power expenditure due to suction line velocity, as well as the reduced capacity due to lower suction pressure, for every $1.00 power cost in the first instance we have $1.09 power cost in the second. Condensing unit maintenance costs will increase proportionately on account of increased running time.

And this is not the end. Referring again to the graph, the evaporator temperature is noticeably higher in the second case. If the system is one requiring close or accurate humidity control, this control will not be accomplished on account of this higher evaporator temperature. This is true even under offpeak load conditions. The condensing unit and the evaporator may handle the total load at these times, but the proper relative amounts of sensible and latent heat capacity will not be maintained.

The ability of a coil to remove moisture depends entirely on the difference between its temperature and the dew point temperature of the entering air. Since this temperature difference is affected to a greater degree than the sensible temperature difference, the latent heat capacity is adversely affected to a greater extent than the sensible capacity. It is almost an inherent fault of the average comfort air conditioning system that it maintains excessively high humidities during periods of low sensible heat loads. The condition described above serves to aggravate this inherent weakness.

Further analysis demonstrates that an additional reduction in suction line size produces a disproportionately larger reduction in capacity and increase in running time. Reference to the chart on page 80 of the March issue shows that this Btu load in a $1\frac{1}{8}$ in. OD suction line produces a pressure drop of approximately 20.0 psi per 100 ft length. This results in a total pressure drop, including the heat exchanger, of 31.0 psi. Points *G*, *J*, and *H* of the accompanying graph will then represent the operating conditions of this extreme case. An analysis of the suction vapor as before shows that considerable power is expended in overcoming the increased resistance.

*Heat exchanger end*
$P = 39.0$ psig
$T = 65$ F
$H = 86.32$ Btu per lb
*Compressor end*
$P = 13.0$ psig
$T = 65$ F
$H = 87.31$ Btu per lb

Heat gain in the suction vapor is then 0.99 Btu per lb, or $0.99 \times 3020 = 2990$ Btu per hr.

Converting to power, $2990 \times 0.000393 = 1.175$ hp, or $2990 \times 0.000293 = 0.875$ kw.

This represents about 6 per cent of the total motor horsepower. The accompanying graph shows the condensing unit capacity at this suction pressure to be 152,000 Btu per hr, a loss of 48,000 Btu per hr or 24 per cent of the original capacity. The system is then undersized by 24 per cent during peak load periods. It will be required to operate 31.6 per cent longer over any given period of time and the operating cost will be 31.6 + 6.0 or 37.6 per cent higher. For every $1.00 power cost in the first installation the power cost will now be $1.38, maintenance costs increasing accordingly.

The cost of the suction piping, together with control and shutoff valves and all other fittings, is a small part of the total cost of an installation. In very few cases does it exceed 10 per cent. It is certainly not possible to effect a saving of more than one-third of this cost by a reduction in the size of the piping. Almost any saving would consist of reduced material costs, the labor of erection remaining substantially the same. It follows then that about the maximum saving that could be effected would represent about 3 per cent of the cost of the complete installation. It has been shown that an effort to accomplish this is false economy, since by so doing the power cost, the operating time, and therefore the maintenance costs can be increased by from 6 to 38 per cent.

It is therefore of considerable importance that the layout and installation engineer consider the suction line as a definite component part of the installation. Its proper selection may—and very often does—have as much bearing on the operating characteristics and on the success or failure of the installation as does the selection of an evaporator or a condensing unit.

## FLOUR MILL USES ELECTRIC UNIT HEATERS

When officials of the Mountain City Mill Co., Chattanooga, Tenn., put in their new plant heating system, they found a way to cut heating costs appreciably, control temperatures better throughout the plant, and make more space available for productive purposes. They also helped to prove that in areas where power rates are comparatively low electrical space heating is worth consideration.

Company officials first experimented by heating several areas of their plant—a large flour mill—with all-electric unit heaters. As a result of this preliminary test, the mill's old heating system was removed and 50 such units, with thermostatic control, were installed.

The main mill has five floors, with insulated ceiling and tight-fitting windows and doors. The mill was heated 24 hr a day, seven days a week, during the winter months. The connected load for the mill's 598,650 cu ft is 655.5 kw, with 50 heater units ranging in capacity from 7.5 to 20 kw. It was found that this load could be operated for a large part of the time off the peak of the 900 kw demand of the mill's productive equipment.

The electrical rate the company pays is 0.004c per kwhr.—H. J. LINDSAY, Electromode Corp.

## RECOMMENDS FIRE VENT FLUES AND DUCTS

Recommendations designed to minimize loss of life in hotel fires made public by the executive committee of the Building Officials Conference of America, Inc., 51 E. 42nd St., New York 17, following review of findings and conclusions of the conference's committee on injuries into disasters based on an investigation of the Winecoff hotel fire in Atlanta in December, include, among other things, the elimination or sealing of transoms in hotel guestrooms; the protection of all floors by supervised watchmen's services, in lieu of adequate fire protection equipment provided with automatic transmission of alarm to the fire department; and the installation of fire vent flues and ducts to conduct superheated air and gases to a point of dissipation outside buildings.

# FOUNDRY VENTILATION

## *Jim Black and Les Avery Suggest How to Plan for Good Conditions*

***GOOD AIR in foundries starts with the original planning of the foundry, and the authors (special engineer and president, respectively, of the Avery Engineering Co.) wind up this article with several specific suggestions of the things to be considered and what to do about them. Before that, they discuss the problem as applied to a molding line in a mechanized grey iron foundry having seven such lines. Previous articles on foundries by the same authors were published in March and April.***

ENGINEERING THE supply and exhaust ventilating systems for a foundry is quite a job. For example, let's consider the No. 5 molding conveyor line in a mechanized grey iron foundry having a total of seven lines. This line turns out the most weight of metal per day—about 85 tons in 8 hr. The casting is a six cylinder engine block and the patterns are ingeniously fashioned to make two blocks in one mold. The total weight of the casting is 500 lb; there are 100 lb or so of cores inside it.

The molding conveyor is about 166 ft long and is shaped like a racetrack; the long straightaways are only 14 ft apart. Pouring is at one end, final shakeout at the other.

Pouring is done from "inside the track"—that is, between the two straightaways, near the first turn. This doesn't appear to be any particular problem and in one way it should help us. We know that the conveyor must be covered with hoods and tunnels—that will help to partition off this pouring station and keep out the stray breezes and air currents that give trouble. That much we know—that an isolated area is easier to ventilate than one which is wide open to other areas. Besides, the length of the pouring station is only 46 ft. The pouring hood was constructed with a back plate down to the floor. Exhaust capacity was 9200 cfm.

Immediately after the pouring station comes a full covering tunnel. The tunnel goes around the turn and ends at the sprue station. This closes in the pouring area completely and the men work with their backs to the metal wall of the tunnel (no drafts).

A supply duct was placed above and behind the men, handling 9200 cfm of unheated outside air passed through them in the direction of the hood. You'd think 9200 cfm of ventilation air would be generous for just four men, but this pouring area was murder even in mild weather. With one 1000 lb ladle of 2700 F iron being poured and a second one waiting, the heat reflected back and forth between the hood and tunnel so fast you could almost smell it. To get relief from reflected heat the men demanded that the back panel be taken off the hood. This would make the hood ineffective; but the men would rather eat smoke than roast in the heat. No amount of coaxing or reasoning could keep that back panel on.

### Exhaust Increased 10 Times

So the hood was redesigned with 10 times the original exhaust fan capacity; enough to be effective without a back panel. There are now four propeller fans serving four sections of the hood. Only three of them are run most of the time, as the entire 46 ft of pouring station isn't often used.

That fixed it fine; everybody was happy. It's a tremendous heating load in winter, but the results seem to satisfy the men. Any good engineer knows we're handling a preposterous air quantity. But how much air *do* you need to cool a man who's standing beside a half a ton of molten iron? How much air is required to offset radiant heat? However little it is, it's tremendous—and no matter how much it is, it's not enough.

The sprue station comes next on this line and required radical treatment. The first effort wasn't cool enough, the second was; again with a high air quantity. This station is unusually hot and smoky, as the sprue weighs over 50 lb and is so hot that it falls apart when it's taken off, and drips pasty yellow iron. The smoke practically explodes from the mold when the sprue is removed. Air quantities required for ventilating this kind of operation are out of this world.

Next down the line is the cope shakeout. The shakeouts aren't so much of a problem—maybe because they're cool.

The next operation down the line illustrates what you can get into in a foundry that wasn't built for a ventilating engineer (as few foundries are). The molding machines were placed so close to the conveyor that there wasn't room for the covering tunnel. It required a hood 50 ft long that enclosed only the top

**Develop a simple, light weight ventilated coverall for foundry workers**

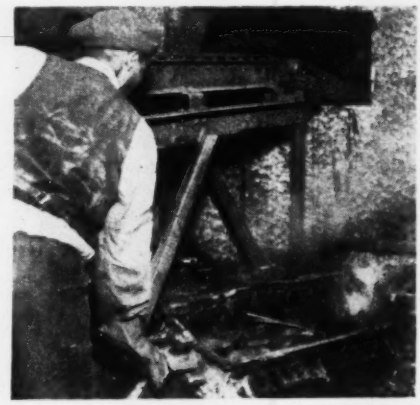
and one side of the conveyor and which of course took five or six times as much air as the full covering tunnel would take. At the next station the drag and castings are removed separately from the conveyor and taken to the drag shakeout, and here again no enclosure was possible since both are removed by an air hoist overhead. Due to the weight of the mold, the length of conveyor required for this operation is 15 ft. There's another 10,000 cfm of exhaust that might be saved by a change in method of handling the casting.

The pouring station on one of the other lines had a still different problem. On this line the casting is a four cylinder automotive cylinder head. It's a small casting, only 50 lb, and the conveyor speeds along at 28 fpm to turn out one head every nine seconds. (When you watch this line turning out heads hour after hour, day after day, hope springs within you that you may get that new car pretty soon.) The whole thing—molding, core setting, pouring, handling, shakeout—is a symphony of coordination and speed and is beautiful to watch.

Anyhow, this little mold has a vent hole in the outside of the flask that spits out gases when the iron is poured in. The gases either catch fire or smoke like crazy. The smoke is squirted *away* from the pouring hood and the hood can't catch it. You could turn the casting around (end for end) on the conveyor and the smoke would squirt *into* the hood. But you can't do that because then the sprue would be on the wrong side and the mold couldn't be poured from a conventional ladle. If the *patterns* had been built by a ventilating engineer, he *might* have put the vent hole on the opposite side.

### Recirculate the Foundry Air?

In the original concept of this job we wanted to recirculate the foundry air—although this practice is frowned on and in some states is prevented by law. By using a progressive cleaning system whereby the cats and dogs are screened from the exhaust air first with a centrifugal dry cleaner, then the air is washed and scrubbed with a dynamic precipitator, you could expect reasonably clean air—probably better than that available outside in a dirty industrial neighborhood. Electric precipitation could be used to remove smoke and fine dust. We still think the idea has merit and can be done, but it



**Make hoods and tunnels and shakeouts flexible so that the size of the casting can be varied and the ventilation adjusted to suit**

will have to wait on developments of continuously self-cleaning apparatus arranged as a three stage operation, all suitable to handle the kind of dusts and acid gases and smoke associated with foundries. When such equipment is available the state will approve the method. Who can say that it would be a hazard when by proper cleaning and recirculation you can be sure of clean ventilation air and save these astronomical Btu's required for heating? Most foundries simply cannot afford to heat the air properly even if the boiler were large enough. So they close up doors and windows as best they can and the exhaust systems get very little air, making what is called an "airbound" building.

Another hope was blasted when we found the excessive cost of trying to capture the wild heat from the cupolas which certainly would go a long way to provide the Btu's needed. On this particular job it simply could not be done at a reasonable cost, but it can and should be done when planning a new foundry.

### Suggestions on Foundry Planning

This brings us to the $64 question. Is there no constructive lesson to be learned, is there nothing the air conditioning engineer can bring to this subject to help the owner who is *willing* to do a job and starts from scratch with a new building? We list a few suggestions (very timidly and very humbly) knowing full well the derisive sneer we will get from a good foundryman when we suggest the start is in the work plan of the foundry! Will someone offer valid reasons why you can't expect a little more cooperation from the production engineers? The ventilating engineer has to be well versed in air conditioning, and then must make a serious study of foundries and foundry practice. The plant engineer and the foundry superintendent should likewise make a serious study of air conditioning. Then you *can* condition a foundry.

Start with the mold design. Plan it for easy pouring under a close hood. Locate the vents to vent into the hood. Make the pouring hood nonreflecting. Separate the pouring station from the balance of the foundry with asbestos board partitions, keeping all molten iron operations in one zone in which a high rate of ventilation is provided with washed clean air. Probably evap-

**Throw away air that has a high contamination or high heat content**

orative cooling would work wonders in this section.

Get rid of the sprues mechanically without having to pile up in a cart. Get rid of hot castings by placing them on a conveyor which immediately takes them outdoors to the cooling tower. Wherever hot castings are conveyed inside the foundry, keep them enclosed in tunnels.

Work out downdraft shakeouts with air supply directly above to drift the clean cool air through the workers. Never, never permit local fans to blow at any working point; they may appear to give the man desired relief but will destroy the planned pickup of smoke and dust and create a *real* health problem.

Plan and locate cupolas so as to be able to use the waste heat in winter when needed and throw it away in summer when not needed. In order to control this heat we suggest that steam or hot water heat exchange is preferable to a direct hot air exchange.

Throw away the air that has a high contamination or high heat content. Clean and recirculate when desired the large masses of air that are used as conveyors and have little contamination. A judicious balance between exhaust and recirculation is going to be worth a lot of money.

Plan an auxiliary air supply system, probably evaporative cooling for summer use. There is no need to tolerate the excessive heat of foundries in summer when the apparatus now available can be used so easily and inexpensively. But in winter you cut back to the minimum air handling, just enough to remove dust and smoke.

Develop a simple, light weight, ventilated coverall for foundry workers which will reflect radiant heat. Probably something in glass cloth will be the answer, white. Yes, this ideal foundry is going to be so clean that the workers can wear *white* clothes.

Make hoods and tunnels and shakeouts flexible, insofar as possible, so that the size of the casting can be varied and the ventilating equipment adjusted to suit. But what is more important, when the work is changed and the hoods, etc., do not fit and can't be made to fit, don't be afraid to design and build a new one which *will* fit the work. In foundry work especially it is not efficient to try to design the air conditioning to meet every possible contingency which may arise in the future. Design for the job in hand, allow reasonable flexibility, and if there is a major change in work—make a major change in the air. Consider the size of the casting, the number of castings per hour, and the total tonnage. If we had time, we would like to prepare a suggested table of air weights as against iron weights in different classes of work. That can and should be done by someone who will get together data from a number of similar jobs. Who will do it?

Foundries will have to be air conditioned, in the broad sense of controlling the temperature, purity, and air motion. Here is truly a great challenge to our profession and our industry. Let's accept it!

# BIG DRAFTING ROOM AIR CONDITIONED

MORE THAN 150 designing engineers in the recently enlarged Cleveland district drafting room, adjacent to the Austin Co.'s general offices, have found that the room's new lighting and air conditioning materially reduce the strain of their work, says the company.

Recessed, fluorescent troffer units on 3 ft centers, extending the length of the room between the welded rigid frame arches, have each been equipped with one daylight and one soft white tube of 40 watt intensity. With mounting heights which range from 13 ft near the outsides of the 70 ft span to 22 ft at the peak, a uniform, well-balanced lighting intensity of over 90 footcandles is maintained on the drafting tables. The troffer units have been suspended from the purlins; and the metal acoustic ceiling is supported by these fixtures.

Three lines of air conditioning supply ducts are located on each side of the room toward the outside of the span. These extend the length of the building and are supported on the rigid frame arches, being located between the purlins. A space between the four lines of purlins in the high center section serves as a return air duct.

Air conditioning machinery, housed in a small two story equipment building adjoining the drafting room, includes a fan room designed to handle 35,000 cfm. A type of electrostatic filter which applies an electrical charge to filter paper has been installed on the second floor across the entrance to the plenum chamber. Reciprocating refrigerating compressors of 100 ton total capacity have been installed on the first floor, with evaporative condensers and other equipment.

**New lighting and air conditioning reduce strain in this enlarged drafting room**

# Design Engineer Reports on Operating Experience

Part of old Panama wall, looking toward the sea and the French Plaza

***John H. Clarke Writes About "Practical Piping Problems"***

BY THE TIME we reached Balboa at the Pacific end of the Canal it was dark, and we puttered around for awhile trying to find a place to anchor for the night, as the repair docks were all occupied. Even though the war was well over, there was still a very considerable amount of shipping going through the Canal — mostly transports, supply ships, and tankers. Many of these ships stopped over for a few days at Balboa; this kept the repair facilities there well occupied. Because of the nature of our repairs we spent nine full days at Panama and were thus able to see a bit of the Canal Zone as well as Panama City.

To a newcomer, this area was very interesting. It was understood that the entertainment and souvenir trade there had fallen off somewhat, but the streets still seemed to be perpetually crowded with throngs of people. A bank and several of the stores there were air conditioned, which made the shopping definitely better in that hot climate. The Eastman Kodak store was a particularly nice stopping-off point in this respect and you could even buy some film there, an added attraction.

While Panama City has many tenement and rundown slum areas, it was really surprising how many modern hotels, stores, restaurants and fine apartment buildings had been constructed in recent years. Fortunately, I had the good services of my roommate, the electrician, who had been through several times before on previous trips, so we were able to make a planned tour of the area and thus see more of the interesting features.

The original site of Panama City was about seven miles down the coast from the present location. But after Henry Morgan sacked and burned the city in 1671, the Spaniards relocated it on a rocky promontory at what is now the Pacific end of the Canal. The old city as relocated was completely walled in and much of this wall remains today, rambling through the various parts of the city and turning up in the most unexpected places. At the extreme end of the promontory is a "U" shaped section of the old wall which now forms a rather popular promenade. From this vantage point we were able to obtain a beautiful view of the sea and of the ships coming and going, and an even more beautiful view at night of the phosphorescent waves rolling in on the beach. This is indeed a sight to see. Inside the wall and at the base of the "U" is a court called the French Plaza, dedicated to the men responsible for the building of the Canal. Along the wall on one side are the old cells, formerly used for political prisoners but now used for garage space and storage.

Incidentally, the Canal area is a particularly good place to observe the phosphorescence mentioned above, because of its considerable brilliance. At night the tugs and launches can be seen traveling to and from the docks and ships with gallant mustachios and turbulent wakes of the bright bluish water. It gives the scene quite a fantastic atmosphere.

While we had ample time in port to do sightseeing, we were also very busy with repairs and routine cleaning and painting. As soon as we had moved to the dock, the plant

***FIRST-HAND experience on how designs perform in service, how the equipment is actually operated, and what troubles are encountered were Mr. Clarke's objectives when he (a design engineer by training and profession) shipped on a tanker as junior third assistant engineer. The experience he got was most illuminating, and he reports on some of the piping and other difficulties here, continuing his tale from our previous issue . . . His comments and observations are of interest not only in connection with marine work, for many of them apply to mechanical equipment in plants and buildings as well . . . His seagoing phase completed, the author is now a design engineer with the Standard Oil Co. of Indiana***

was secured and for the duration of the stay we were provided with electrical power from a shore connection. No shore steam was available so we had no hot water. Quarters heating was decidedly unnecessary.

The broken extraction valve was immediately removed for repair, and the burned-out fire pump motor was removed and a new motor installed in its place. We were very fortunate in this respect because that was the only spare motor of that size and type that they had there; otherwise, it would have been necessary to rewind the motor coils.

The feed pump turbine and pump bearings were checked and replaced. A short run was made to check the pump and bearings and inside of 15 min the bearings had again wiped, necessitating their removal and reinstallation. The job was not completed until just before sailing time, and the pump was not tested to see if it would operate properly. There were several items worth noting in regard to this trouble. In each case that the bearings failed, both the turbine *and* the pump bearings failed. A "fast" flexible coupling was provided between the units. But at no time (to my knowledge) was oil ever provided for this coupling during the six months that the ship was out after leaving Baltimore.

In the boiler room, the firemen and wipers were busy cleaning the boiler burners and registers and painting the boiler room fidley. Elsewhere, there were other miscellaneous small jobs under way, such as repacking a few valves and cleaning the scale out of the drain and blowdown line from the evaporator. We had trouble keeping the latter from plugging up during the trip because of the heavy scale peeling from the coils. Unless the evaporator was thoroughly and frequently blown down (as it should have been), the scale would accumulate over the drain and plug it up solidly. The best method seemed to be to blow the unit down, cold shock it to remove the scale, and then blow it down sufficiently to get ride of the scale that cracked off.

### Expansion Joint Repairs

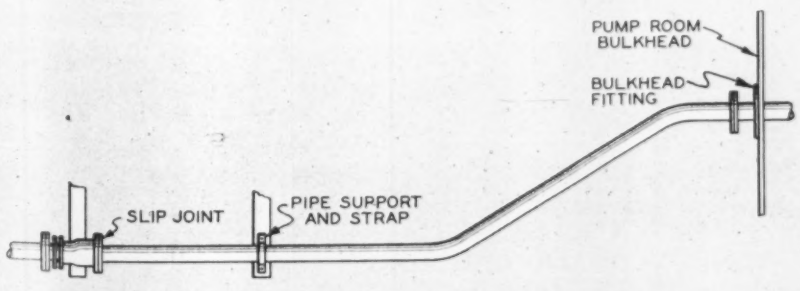
On deck, the shore crew was busy with a job that was to be the beginning of a number of breaks or leaks arising from troubles with piping expansion. In lieu of expansion loops, all of the deck piping was provided with the packed type of expansion or slip joints. On the way down from Baltimore, the aft joint on the 125 psi gage deck steam line (there were two joints between the deckhouses aft and one forward) had begun to leak. By the time we arrived at Panama it was leaking profusely.

Investigation showed that the sleeve was so badly pitted and rusted on the outside that it could no longer operate properly. This had also made it impossible to keep the packing tight and still get any slip in the joint, thus causing the steam leak. When it was removed for repair, it was discovered that the support was about 10 deg out of true. The expansion joint was bolted down hard onto its support by means of four bolts and, for this reason, was under serious stress from the support. Considering the installation and the condition of the piping, it is possible that the support was originally true and had been pulled out of position by the piping stress. All of the slip joints on the ship were severely rusted and pitted so that the joints and piping were not free to move as they should have been. The above slip joint was repaired by providing a copper sleeve and brass flange, this being the only suitable material available of the required size.

The slips on the slip joints were supposed to be of stainless steel but I am not sure that this material was used on this particular ship. Later investigation of another ship known to have stainless steel slips showed that the same rusting and pitting was taking place, although it was not in as advanced a stage because the ship was quite new.

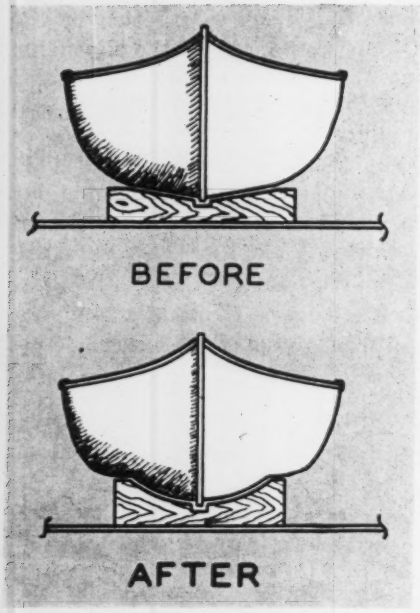
Another factor having a bearing on this pipe expansion problem was that pipe straps were used to hold the piping to the supports, and in every case the straps had been fitted too snugly and had rusted and frozen to the piping, thus making movement at the supports virtually impossible. It was probably this, combined with the rusted condition of the slip joint, and the working of the ship structure and the pipe supports as the ship alternately hogged and sagged that caused the serious bending of the slip joint support mentioned above. This would indicate to me that the deck piping should have been provided with expansion loops in lieu of the slip joints, and that roller guides or some similar means other than pipe straps should have been used to permit free movement of the piping. As it was, continual damage resulted to the deck piping, and twice during the trip the catwalk between the 'midship and aft deck houses which was carried on the pipe supports (the pipe supports were on the catwalk structure), broke loose at the 'midship end, despite two freely moving expansion joints provided in the catwalk.

**The 4 in. deck steam line at the aft slip joint. Note that when expansion or contraction took place from temperature changes or movement of the ship, the offset in the pipe would tend to make it pivot around the support. This was probably a contributing factor in the failure of the slip joint, as any pivoting would cock the slip in the slip joint.**




### Ventilator Screens Clogged

In surveying the ship, our port engineer at the Canal had noticed that the screening over the mushroom intakes to the ventilation heating system for the aft deck house had been removed; the previous crew had done this because of a desire to get all the air possible for their quarters. They had also




**The new chocks didn't fit the lifeboats; that is, they didn't until the boys took up on the turnbuckles. "Before" shows a boat in proper fitting chocks, and "after" shows it reshaped after being pulled down on the improperly fitted chocks**

removed the screens from several of the washroom exhaust outlets because some uninformed person had provided insect screening in addition to the usual 16 ga, ½ in. mesh screen. The insect screen had, of course, filled up solidly and practically stopped the ventilation.

The port engineer seemed to feel it his duty not only to replace the screen, but to provide ¼ in. mesh (about 3/16 in. open) for the purpose. He explained that his reason for doing so was a Coast Guard regulation requiring that mesh for antisabotage.

I was aware that such a recommendation had existed, but new ships were supplied with ½ in. mesh because it had been clearly demonstrated that finer mesh filled up too rapidly with soot, lint and dirt, and in addition, it was felt that the ½ in. mesh offered adequate protection for sabotage. The intakes for *this* system terminated in the fan room on the weather deck. These ships needed all the air they could get and even if a rule *did* exist in this regard, he could have overlooked a point or two considering that the war was over and that *all* of the dozens of other inlets on the ship were provided with ½ in. mesh screen, if they had any screen on at all. (A few of them still did not.)

In the storm off Cape Hatteras coming down from Baltimore several of the life boats had become loose and had taken quite a pounding, so these were removed from the ship, repaired and put back, together with new chocks. The deck crew was told to see that they were properly secured, this time. But someone made a slight mistake. The chocks didn't fit the life boats—that is, they didn't until the boys took up the turnbuckles. After that they had a few extra wrinkles in them that the original designer never intended to provide.

### Extraction Valve Brazed

The extraction valve which was removed from the ship was repaired by brazing the yoke to the bonnet flange. At the same time, the stem was replaced so as to provide a right hand closing valve in lieu of the illegal left hand stem which had been on the valve. It was understood that on the previous trip the original stop valve had been replaced because it was leaking, and the only valve available had been a left hand valve, which was installed. A large sign was provided to indicate the fact.

As soon as the repaired valve was reinstalled, we started up the boilers and got the plant ready for operation. This was a fairly easy task in this port because shore power was available, and we could get diesel oil from shore to get the burners started. A small tank of diesel oil, a portable pump, and about 75 ft of copper tubing was used for the purpose, the latter being connected directly to one boiler burner manifold. The later ships of this class were provided with 75 kw diesel generators and a 500 gal diesel oil tank for cold ship starting. All we had was a small kerosene tank for starting fuel, but no source of power for the fans or any of the other auxiliaries.

# HOT AIR AND COLD AIR FROM SAME PIPE

A "HOT-COLD PIPE" 15 in. long and 1 in. in diameter that converts ordinary compressed air into both hot and cold air without the aid of moving parts has been built by Westinghouse research engineers. As yet scientists have no explanation for the phenomenon, though it is conjectured that the cause may be a frictional effect between gases moving at different velocities. To study it, a working model has been made.

In operation compressed air is pumped into a nozzle at one end of the pipe. There it strikes the inner wall of the pipe tangential to the diameter and is converted into a whirlpool of spinning gases. The air in the center becomes cold almost instantly while that on the periphery gets warm. The cold air is drawn off through a ¼ in. opening at the center of one end, while the warm air is tapped from the periphery.

With the original tube 3/10 in. in diameter, R. Hilsch, its German inventor, claims to have produced simultaneous jets at 154 F and 10 F. The present model has purposely been constructed larger in diameter to expedite instrumentation for scientific study. For this reason it gives a temperature difference of only several degrees.

Low efficiency rules out commercial application at this time, but like the "blue glow" in Edison's experimental electric light that led to his invention of the first electronic tube, it may lead to unexpected developments. Even now it holds promise as a handy laboratory tool for making cold as simply as a bunsen burner makes heat.

*Question of the Month*

# Practical Suggestions on Unit Heater Control

IN THE MARCH HPAC, "J.B." asked for the following information: "The usual method of controlling a steam unit heater is to have the fan on or off according to the demands of a thermostat located in the area heated. When the fan is off, however, with steam pressure in the heater, a fair amount of heat is still emitted from the unit—so on some jobs there is a valve in the supply line to the unit to turn the steam as well as the fan on and off. I would be interested in any information on the relative advantages and disadvantages of this more elaborate and therefore more expensive method of control on unit heater installations."

Following are some of the answers to "J.B.'s" question.

## HEAT OUTPUT WITH FAN OFF NEGLIGIBLE

The question regarding use of valves on unit heaters is excellent, being of the variety which those concerned with the manufacture of unit heaters, for example, tend to overlook because they are too familiar with its substance. Doubtless this question has troubled others than "J.B."

In practical terms, the valves on unit heaters are there to shut off the unit heater branch line for servicing of the traps or possibly the unit heaters. In only one case out of several hundred is a unit heater valve likely to be accessible without the use of a ladder and it is thoroughly impracticable to turn this valve on and off each time the unit heater motor is switched on or off.

The heat dissipation from an average unit heater with the fan not running is on the order of 4 to 5 per cent of the operating capacity of the given unit. This is so small a fraction as to be fairly negligible. True, the heat emanating from a unit heater which is not running will rise to the ceiling and pocket there in a presumed uneconomical manner. It can be remembered, however, that a substantial portion of the total heat loss from a single story structure in which unit heaters are employed is through the roof. This being the case, there is a continuous forming of cool air along the ceiling and the modest bit of heat rising from a nonoperating unit heater can be valuable as an offset, to prevent this chilled air from dropping onto room occupants. It is true that a corresponding recovery is not effected in other than the top floor of a multistory building, but even in these lower floors the wastage of heat is small, inasmuch as such heat accrues partly to the benefit of the floor immediately above and inasmuch as intermittently the temperature gradient is restored by the units going into operation.

The future usage of unit heaters will involve greater employment of temperature modulation means of one or another kind, but this will be aimed not so much at increasing the already high operating economy of unit heaters as it will at a further step up in the degree of comfort provided workers in factories and other spaces served by unit heaters.—A. G. D.

## SEVERAL METHODS TO SOLVE THE PROBLEM

There are several ways of solving "J. B.'s" problem, depending upon the particular application of the unit heater. I have used the following methods with satisfactory results:

One recently completed job (which might not be considered good practice but is nevertheless functioning satisfactory) had a small unit heater in a rather small office space with a low ceiling. This unit heater takes care of infiltration caused by the opening and closing of an outside door. It happens to be connected to one end of a thermostatically controlled steam line which is handling direct radiation in about 15 small offices whose heat losses are about equal.

The thermostat controlling this steam line is located in an office remote from the unit heater. It was not advisable to run a separate steam line to serve this unit heater due to its location, which would have made a rather costly installation. In order to eliminate a cold and objectionable air stream from the unit heater fan when the steam was cut off we installed a pressure switch in the steam line adjacent to the unit heater and calibrated the pressure element low enough to utilize all available heat in the unit before cutting the fan off.

I have used this method in other applications of unit heater control and it may be one answer to the problem of "J. B." He may use one pressure switch (preferably near the end of one of the longest runs) to actuate a small relay for each unit heater and control a number of unit heaters (assuming they are not too far apart). This would

***You are invited to submit a heating, piping, or air conditioning question for publication here. You are also invited to submit answers to published questions, which are paid for—if suitable for publication—at regular rates with additional payment for sketches. Address the Editorial Department, Heating, Piping & Air Conditioning, 6 North Michigan Ave., Chicago 2, Illinois.***

merely be a matter of which was the most economical—electric wiring and relays or a separate pressure switch for each unit.

Another method I have used with success, especially in areas where we have a high ceiling and our heating units are mounted correspondingly high, is as follows:

We use two thermostats, one in the area of occupancy to control the steam supply to the unit heater. The second thermostat or its capillary tube is located in the vicinity of the heater and it controls the fan motor. If the motor is fractional hp and there is no relay in the circuit, a high voltage thermostat can generally be used more economically. The fan will continue to run as long as there is any warm air in the vicinity of the thermostat, the desired cutoff temperature setting of the thermostat being determined empirically in the field. This method has the added advantage of pushing the warm air down to the area of occupancy, which is usually advisable with a high ceiling as there is quite often a considerable temperature differential between the floor and ceiling.

We also have areas where no discomfort is caused by letting the fan run constantly and thermostatically controlling the steam supply only.

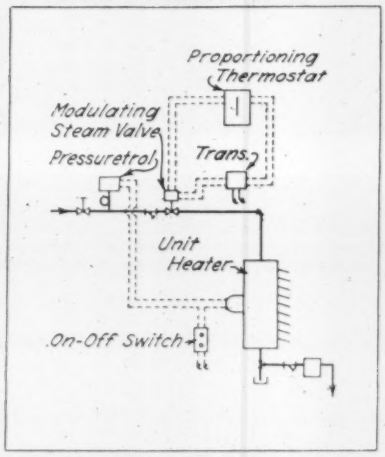
On any installation where the fan of the unit heater is used to create an air movement, during the season when no heat is needed, it is usually advantageous to install a "summer switch" which will shunt out the pressure switch or thermostat so that the fans may be run constantly or as desired, regardless of the position of the pressure switch or thermostat.

I hope the above will be of some value to "J. B."—W. B. P.

### MODULATING CONTROL HAS ADVANTAGES

In reply to "J. B.," it has been my experience that modulating control of the steam supply has definite advantages over on-off fan control and on-off steam control.

With the use of a modulating steam valve operated by a proportioning thermostat, the fan operates continuously and the steam supply to the unit is throttled. With this method a constant discharge temperature for a given room condition is maintained and the hot blasts of air are eliminated, making for better breathing level comfort.




**Modulating control has definite advantages, says "N.F."**

Also, with the fan continuously running, the tendency toward stratification, as is the case with on-off systems, is eliminated.

The only other control we would use in connection with the preceding would be a pressure instrument connected in series with the fan motor, so that the fan would not operate without pressure in the steam main.—N. F.

### THERMOSTAT'S ALL THAT'S NECESSARY

Unit heaters give more comfort when installed in strategic places (no draft) and at the proper height; room thermostat control of the motor, to my mind, is all that is necessary.

The small Btu emission from the unit when the fan is not operating can be charged to ventilation, which is just as important as bodily heat requirements.

The fan is stopped and air throughout the plant begins to balance itself—gases and bodily odors will rise, encouraged by the small amount of heat from the units, going to the ceiling and usually out into the atmosphere through ventilating hoods.

I do believe the fan stoppage is really important; it gives the bad air a chance to rise well above the fan and not recirculate continually; also, should the steam valve close with the motor stoppage, the cool air blown down on the occupants when again starting would be most uncomfortable, as the fan will attain its full velocity before the heater is fully circulated and delivering heated air.—J. J. F.

# Preventing Condensation on Store Windows

DESIGN OF A MODERN confectionery store with a front almost entirely of glass presents the problem of how to prevent condensation on the glass in cold weather, according to "W. L.," in a question published in the February HPAC. There will be no show window back. The glass extends down to within 8 in. of the floor.

Double glass was considered but it cannot be obtained in the required size. The interior of the store will be air conditioned, which means that a certain amount of humidity will always be present.

"W. L." asked for suggestions on how to meet this problem, and several were published in April. Additional answers are given here:

### USE RADIATOR, FAN, OR UNIT HEATER

The solution to "W. L.'s" problem may be found in one of two methods or a combination thereof. The first method is to keep the inside surface temperature of the glass above the dewpoint of the atmosphere of the adjacent inside air. The second is to create rapid air movement over the inside glass surface so as to evaporate the moisture being condensed thereon faster than it can build up a visible film.

One means of effectively using Method 1 would be to put a high capacity heat radiator (low slung design, of course) along the lower inside sill of the window.

A means of utilizing Method 2

would be to locate a high velocity fan on an adjoining wall or on the ceiling so as to play a fast moving current of air over the window area.

A combination of the two methods could be provided through the installation of a unit heater near the ceiling so placed as to direct a warmed air stream down against the glazed area.

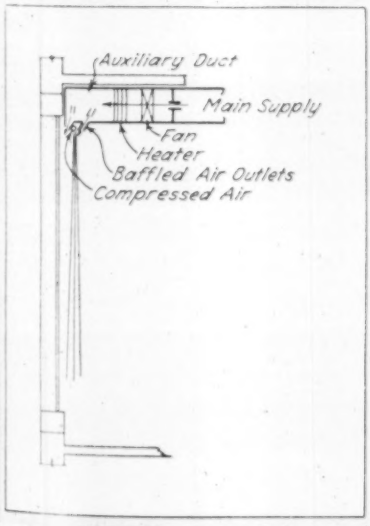
The capacity of any of the systems mentioned could be determined by any good heating and ventilating man. It is hoped that the foregoing suggestions will prove helpful.—R. A. B.

## BLANKET WINDOW WITH AUXILIARY AIR

It would be possible and seems to be simplest to adjust inside air conditions in such a way as to stop condensation on windows. However, this would involve an unusually low relative humidity which could be obtained only with an air conditioning plant which would be more expensive in investment and less economical in operation than a more conventional plant. For this reason special means should be provided.

The problem is to keep the temperature of the inner surface of the glass higher than the dewpoint of the air close to the windows. To accomplish this, the film coefficient for the air on the inside should be high and a high air velocity is required. Further, the dry bulb temperature of the air should be high and its relative humidity should be low. These conditions can be obtained by branching a duct off the delivery duct of the general conditioning plant and arranging an additional air heater and an additional fan in the branch. Such an auxiliary duct should run on top of the glass front with the air flowing down the glass.

**"M.H." suggests blanketing the window with compressed air**




Assuming air in the general supply duct at 70 F dry bulb temperature, 50 per cent relative humidity, 51 F dewpoint, being reheated to 90 F and leaving the glass surface near the floor at 80 F without loss or gain of moisture, the inner glass surface should be maintained at a temperature of 51 F or more, and this is particularly required for the lower zone, where the air temperature reaches 80 F.

With strong outside wind, as much heat as 3 Btu per hr per F per sq ft might be transmitted from outside to the inner surface of the glass. Thus, for 0 F minimum outside temperature, 153 Btu per hr per sq ft would be transmitted if the inner surface reaches the admissible minimum temperature of 51 F. The velocity of the inside air flowing down the glass should then be such as to transfer 153 Btu with a temperature difference of 80 — 51 or 29 F, or 153/29, or 5.3 Btu per hr per F per sq ft. A velocity of about 20 fps should answer this requirement.

Such velocity could not be maintained without special means, as the throw of the air leaving the overhead duct would not suffice. Moreover, excessive quantities of auxiliary air would be needed. The problem can be solved by using air blankets between the room and the auxiliary air stream. Small pipes delivering compressed air would be arranged parallel to the auxiliary air duct with numerous openings through which the air escapes at very high velocity in a direction parallel to the glass surface, restricting the flow of the auxiliary air to a narrow path. Such blankets will entrain air from both sides. Therefore, they should be erected at some point within the auxiliary air stream in order to prevent air from the room being entrained on the innermost side of the blankets. The quantity of compressed air required depends on the height of the glass front. For average conditions 2 cfm free air compressed to about 20 psi gage pressure would be required for each 1 ft run of the glass front.

The sketch shows the described arrangement.—M. H.

## WANTS DATA ON SIZING HIGH PRESSURE RETURNS

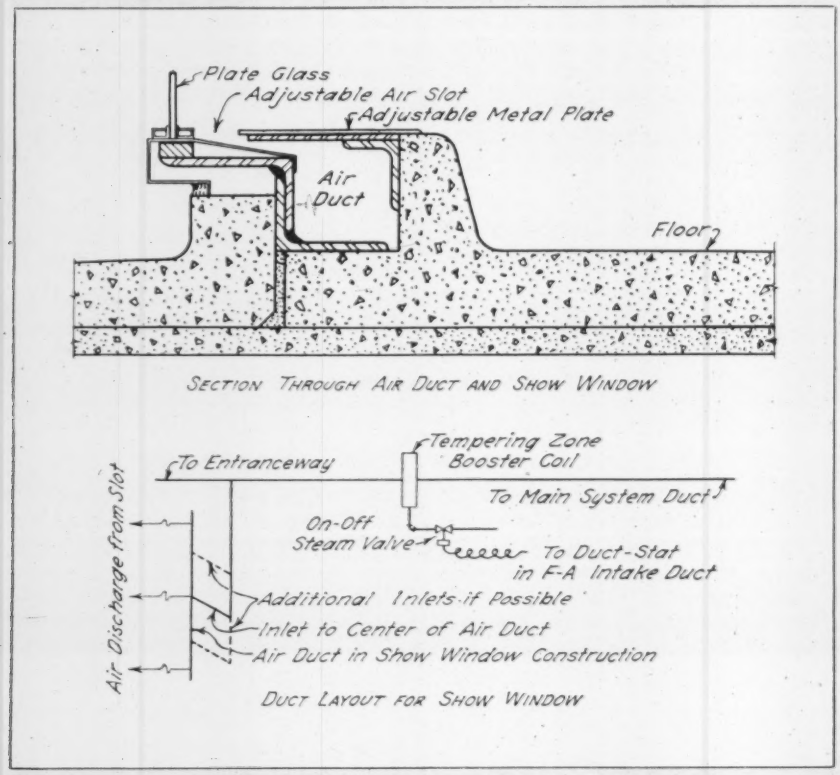
"H.M.S.," an operating engineer reader of HPAC, would like information on sizing of high pressure return piping. He writes that the ASHVE's Heating, Ventilating, Air Conditioning Guide contains some data but that it is brief and deals only with systems operating at 30 or 150 psi. He will welcome comments on this subject of sizing return piping, references to sources of information, etc.

## I HAD THE SAME PROBLEM TO SOLVE

"W. L.'s" question about the store window interested me very much because I had to contend with the same problem some time ago and wish to present the solution for what it may be worth.

The store in question also was a confectionery combined with a restaurant and completely air conditioned. Since it is common practice to temper the cold air entering through the entrance into store buildings in the colder regions of this country, it was decided to utilize the same blast coil and zone to furnish hot air for the defrosting of the show windows. Because the temperature of the air used for tempering in the entranceway is higher than that delivered into the sales space for heating purposes, the prevention of condensation on the show windows is aided by requiring a smaller volume of hot air for this purpose.

The accompanying sketch shows how the inherent construction of the show window itself was utilized for the distribution of the air. The width of the slot was left undetermined until the system was placed into operation, at which time it was adjusted to provide even distribution over the entire length of the window. The air duct as shown on the sketch will never be very large and thus will generally cause a high static loss for any large amount of




**"R. J. H." solved the same problem with this scheme**

air that is moved through it. This static loss can be reduced by providing several inlets into the air duct and thus reducing the volume of air through any section of the duct. All too often, however, it will be impossible to have more than one inlet into the duct at one end or the other. An inlet at the center, or one at each end, will have the effect of doubling the carrying capacity of the duct, and materially simplifying the balancing of the system.

Best results are obtained by high temperatures and velocities through the slot. The high velocities are needed to allow the air to reach as high as is necessary to sweep the whole window area susceptible to condensation. The air stream so directed will also help to prevent any direct stream of hot air passing over the displays and melting them.

This system will operate only when the tempering zone is in operation. If separate control is desired, it can be obtained by installing additional ductwork and the necessary dampers. It is not believed that such will ever be the situation unless it is desired to cut off the tempered air to the window and not the entranceway.

The air for this defrosting process is furnished by the main air conditioning system. The main system furnishes air at a temperature of approximately 90 deg at the outlet grilles. Since it is necessary to have a higher outlet temperature for the entranceway, the air is passed through an additional booster coil which raises it to the desired higher temperature. The booster coil is controlled by an on-off steam valve which in turn is controlled by a ductstat installed in the fresh air duct.

It is thus obvious that outdoor conditions will govern the operation of the booster coil. It will also be seen that unless specific provisions are made for shutting off the tempering zone, warm air will always be supplied during the heating season and cool air during the cooling season. The cool air may be necessary during the cooling season to preserve some of the displays. This can be determined for each individual installation under its own operating conditions. Care must be taken, however, to prevent condensation taking place on the displays themselves, possible under the right conditions.—R. J. H.

## GAS SALES RISE SHARPLY

Sales of gas utility companies to ultimate customers in February totaled 2,951,150,000 therms, an increase of 11.2 per cent over February 1946, the American Gas Association reported recently. For the 12 months ending February 28, 1947, gas sales totaled 25,970,812,000 therms, a gain of 3.0 per cent over the comparable period a year earlier. The association's index of total gas utility sales on February 28, 1947, stood at 206.8 per cent of the 1935-1939 average.

Extended periods of low temperatures accompanied by high winds throughout the month increased heating load demands to points that heavily taxed production facilities of all branches of the industry. Degree days in the manufactured gas area were 8.3 per cent over a year ago. In the natural gas areas degree days were 20.6 per cent higher, while mixed gas areas experienced a rise of 14.6 per cent in degree days as compared with February 1946.

## DECORATED FOR WORK ON "OPERATION PLUTO"

Lester M. Goldsmith, chief engineer of the Atlantic Refining Co. of Philadelphia, has been decorated for exceptional civilian service for his contributions to the installation of an oil pipe line under the English channel preceding the invasion of Normandy—the famed "Operation Pluto."

The presentation was made in Washington by Lieut. Gen. R. A. Wheeler, chief of engineers of the U. S. Army. The citation accompanying the medal states the award was made to Dr. Goldsmith "for exceptional service with the Office of the Chief of Engineers during World War II. His outstanding performance in coordinating specifications for and procuring pumping equipment and lead cable essential to installing a petroleum pipe line across the English channel contributed to the successful prosecution of the War."

The award is the highest honor the War Department can confer upon a civilian.

*Sam Lewis' Page*

# Condensation Problems Serious With Indoor Swimming Pools

## *Ventilation and Heating Design Must Protect Structure*

INDOOR SWIMMING pools surrounded on all sides and the top by heated rooms are fortunately situated, for there are no walls or roofs having cool surfaces against which the inevitably high moisture content in the air can condense. In such cases, there generally is no serious deterioration of structure or surfaces due to water. On the other hand, the natatorium often is in a more or less separate wing of a building and may have several outside walls, skylights, and a roof. Condensation problems are then serious.

There is a large municipal swimming pool—some 10 years old—that was covered by a roof and walls of greenhouse construction, with structural steel exposed to the interior moisture and with the glass in steel frames. The room was heated by steam radiators, with open windows depended upon for ventilation. The effect of condensation was so marked that the metal frames for the glass were destroyed by rust and corrosion, and the heavy steel trusses and beams were so damaged that their structural strength was weakened seriously.

Installation of mechanical ventilation and entirely new wood frames for the glass has delayed for a time the ultimate complete replacement of the enclosure, although replacement is eventually going to be required. Even though the radiators were removed so as to require operation of the fans, running the fans and the heating and drying of new air from outdoors cannot be expected for 24 hr every day. Lacking a constant replacement of the damp air by new air capable of absorbing moisture, the moisture precipitates on the cool surfaces, and damage continues.

In another example, the swimming pool of a large high school has a thick concrete roof carried by steel trusses. There is excellent insulation above the concrete under the composition roof surface. Transverse rafters are suspended at the level of the truss bottoms, and an effective acoustical ceiling is fastened to them. There is but one wall of this natatorium exposed to outside temperature. It has double sash. The room has no means of being heated other than by a supply fan, convectors, and ducts that deliver and distribute air that has been heated to a temperature sufficiently warmer than the 80 deg desired in the room so that the room temperature remains constant while the fans operate.

However, there are nights and Saturdays and Sundays and vacations—and during such hours the fans are stopped. It is not practicable to drain and waste the filtered water in the pool for such relatively short periods. Therefore, the water in the pool—warmer than the air in the room—continues to moisten the air.

**High moisture content presents a problem**

### Noisy Place

A natatorium is about the noisiest place one can find, due to the reflection of the shouts of the swimmers and spectators from the water and walls, so that without sound absorbing surfaces somewhere, the room may be most unpleasant. Therefore, the natatorium in this high school, with glazed tile walls and glass windows, was much improved by the noise absorbing ceiling.

*In this regular feature, informal comment on heating, piping and air conditioning matters is given by Samuel R. Lewis, consulting mechanical engineer and member of HPAC's board of consulting and contributing editors. This month he discusses the swimming pool problem of preventing moisture condensation and the resulting deterioration of structure and surfaces. He concludes by suggesting a possible solution for the difficulty.*

The construction of this roof, generally as described, is typical of most swimming pools.

I entered the room on Good Friday morning. The building was unoccupied except for a few janitors and the operating engineer. No fans were running. One of the four boilers had a banked fire. The water in the pool was crystal clear. The temperature of the room was about 70 deg, due to heat storage from the normal operation of the preceding day.

I noticed frequent wet spots on the floor of the bleachers. I touched the wall below one of the windows and my hand came away dripping wet. I observed occasional disturbances in the water of the pool with widening ripples as from drops of water falling from the ceiling. Thereupon, I looked at the ceiling. The acoustical blocks were wet! Some of them had disintegrated to such an extent that their original constituents were shamefully exposed. Several of the blocks appeared about to fall, and as they were made of a composition of decided weight, I know I prefer to be out from under when they finally decide to come down. The attic then required inspection, and the steel up there, despite the insulation on the roof, was deeply rusted and evidently had been wet and undergoing chemical action for years.

### Same Problem in Industry

The same sort of thing, of course, has been happening in industrial establishments where water vapor from water-using processes condenses on roofs and walls. This has been observed and fended against in paper-making plants and in textile weaving mills. It happened in the top story of a hospital where the roof had been designed as an open solarium, insulated below the walking surface, but without positive ventilation in the joist spaces above the top story ceiling and without fire stops in the furred outside walls. In this case, moisture flowing up from a very damp basement condensed between the joists and the water dripped down on the patients' beds through the openings around the lighting fixtures. (I fixed that one and averted some litigation and hard feelings against the insulation people.)

### A Possible Solution

A possible solution of the problem of the natatorium that cannot have warm rooms all around it (few may), and that cannot be built with moisture-shedding interior surfaces (none can), and that must have noise absorbing areas (all should), might be to omit all of the windows and skylights, using waterproof glazed tile walls, and to depend entirely on electric illumination. Mechanical ventilation is, of course, imperative. An attic space and the best possible roof and wall insulation against heat transfer should be provided. Then employ radiant hot water pipe coils in the ceiling, with heat insulation above the pipes, the insulation preferably faced with a reflective surface.

The pipe coils might be something like 12 in. apart over the entire ceiling. They would be imbedded in cement plaster. They would be kept warm the year around (condensation and damage come in summer whenever there is a sudden change in outdoor temperature) only to such a temperature that the ceiling would be warmer than the dewpoint temperature of the air in the room. The cost for heat thus applied is not serious compared with the alternative cost of eventually rebuilding the enclosure of the room.

Any desired type of organic or inorganic acoustical treatment can be applied to the plaster under the warm coils. Automatic control of the water temperature may easily be employed.

I have used this scheme successfully to prevent rusting of costly steel in otherwise unheated warehouses.

# Developing Process To Make Gas From Coal

PITTSBURGH Consolidation Coal Co. has announced plans for a research and development program which it is believed will lead to the creation of a new multi-million dollar fuel industry. The aim of this program is to perfect commercial processes for making gasoline and a gas fuel of high heat value from bituminous coal.

A commercial plant for making liquid fuels and a high Btu gas, which it is hoped might be ready for operation by 1950-1951, could be located within some 25 miles of Pittsburgh, adjacent to one of the company's huge coal tracts. One single large plant of the size being considered could yield a gas of high heat value in quantities equal to the output of the Big Inch and Little Inch pipe lines and create a local source of gasoline, diesel, and other fuel oils and a certain amount of alcohols.

A unique advantage of the projected conversion process is that it has a flexible pattern of production. It could be arranged, for example, to produce 400 million cu ft of gas per day in the winter months and reduce this to half that quantity during the summer months when the plant would be set up to produce larger quantities of gasoline.

# DATA SHEET

## Conductivities and Conductances of Building Materials and Insulations

*By Paul D. Close*

CONDUCTIVITY AND conductance values have been compiled by the author from published sources and are given in Tables 1 to 11 inclusive. It will be noted that the values in Tables 1, 3, 6, 7, 8, 9, 10 and 11 are conductivities ($k$) and are therefore based on a 1 in. thickness of the material reported in each case. Tables 2 and 4 are conductances ($C$) and are for thicknesses other than 1 in. or for combinations of two or more materials. Table 5 contains both conductivity and conductance values. All coefficients express the rate of heat transfer between the material surfaces in Btu per hr per sq ft per deg Fahrenheit temperature difference per inch thickness (where the conductivity is given) or for some other thickness or combination of materials (where the conductance is given). Resistances ($R$) or resistivities, which are the reciprocals of the conductances or conductivities, are also listed, as well as the authorities in each case.

The tests were conducted by generally reliable sources. There is no certainty, however, that the figures for proprietary materials correspond with those for similar materials purchased in the open market. These and other pertinent facts should be considered when using these data.

Tables 1 and 2 were published in January, Tables 3 and 4 in February, Tables 5, 6, and 7 in March, and Tables 8 and 9 in April.

[Note: Mr. Close is technical advisor for Simpson Industries, Inc., sales division of the Simpson Logging Co. He was formerly technical secretary of the Insulation Board Institute, and before that was technical secretary of the American Society of Heating and Ventilating Engineers.]

### TABLE 10—CONDUCTIVITY ($k$) OF BLANKET AND BATT INSULATIONS

*These coefficients are expressed in Btu per hour per square foot per degree Fahrenheit temperature difference per inch thickness*

| MATERIAL OR TRADE NAME | DESCRIPTION | DENSITY (lb per cu ft) | MEAN TEMP (F) | CONDUCTIVITY ($k$) | RESISTANCE ($R$) | AUTHORITY OR REFERENCE |
|---|---|---|---|---|---|---|
| Balsam Wool | Chemically treated wood fibers between layers of strong paper | 3.62 | 70 | 0.25 | 4.00 | (3) |
| Balsam Wool | Chemically treated wood fibers between layers of strong paper | 2.2 | 90 | 0.27 | 3.70 | (1) |
| Balsam Wool | Chemically treated wood fibers between layers of strong paper | 4.8 | 35 | 0.25 | 4.00 | (4) |
| Cabot's Quilt | Eel grass between kraft paper | 3.4 | 90 | 0.25 | 4.00 | (1) |
| Cabot's Quilt | Eel grass between kraft paper | 4.6 | 90 | 0.26 | 3.85 | (1) |
| Cotton | Short staple linters, fireproofed | 6.25 | 90 | 0.245 | 4.08 | (1) |
| Cotton | Short staple linters, fireproofed | 4.50 | 90 | 0.237 | 4.22 | (1) |
| Cotton | Short staple linters, fireproofed | 2.45 | 90 | 0.238 | 4.20 | (1) |
| Cotton | Short staple linters, fireproofed | 1.60 | 90 | 0.262 | 3.82 | (1) |
| Cotton | Short staple linters, fireproofed | 0.85 | 90 | 0.289 | 3.46 | (1) |
| Cotton | Short staple linters, fireproofed | 0.65 | 90 | 0.298 | 3.36 | (1) |
| Dry Zero | Kapok between burlap or paper | 1.0 | 90 | 0.24 | 4.17 | (1) |
| Dry Zero | Kapok between burlap or paper | 2.0 | 90 | 0.25 | 4.00 | (1) |
| Hairinsul | 75% hair, 25% jute | 6.2 | 90 | 0.27 | 3.70 | (1) |
| Hairinsul | 50% hair, 50% jute | 6.1 | 90 | 0.26 | 3.85 | (1) |
| Kimsul | Creped paper stitched together | 1.5 | 70 | 0.27 | 3.70 | (3) |
| Mineral Wool Batts | 4 in. mineral wool batts, barrier lapped on warm side; horizontal position | 3.0 | | 0.31 | 3.22 | (4) |
| Mineral Wool Batts | 4 in. mineral wool batts, barrier lapped on warm side; vertical position | 3.0 | | 0.33 | 3.03 | (4) |
| Mineral Wool Batts | 4 in. mineral wool batts, no barriers; horizontal | 1.77 | | 0.30 | 3.33 | (4) |
| Natur-Temp | Cotton insulating batt | 0.875 | 72 | 0.24 | 4.17 | (3) |

## TABLE 11—CONDUCTIVITY (*k*) OF SLAB OR BLOCK INSULATIONS

*These coefficients are expressed in Btu per hour per square foot per degree Fahrenheit temperature difference per inch thickness*

| MATERIAL OR TRADE NAME | DESCRIPTION | DENSITY (lb per cu ft) | MEAN TEMP (F) | CONDUCTIVITY (*k*) | RESISTANCE (*R*) | AUTHORITY OR REFERENCE |
|---|---|---|---|---|---|---|
| Corkboard | No added binder | 5.4 | 90 | 0.25 | 4.00 | (1) |
| Corkboard | No added binder | 7.0 | 90 | 0.27 | 3.70 | (1) |
| Corkboard | No added binder | 10.6 | 90 | 0.30 | 3.33 | (1) |
| Corkboard | No added binder | 14.0 | 90 | 0.34 | 2.94 | (1) |
| Corkboard | | 8.3 | 35 | 0.27 | 3.70 | (6) |
| Corkboard | | 8.3 | 60 | 0.28 | 3.58 | (6) |
| Corkboard | | 12.2 | —60 | 0.264 | 3.79 | (4) |
| Corkboard | | 12.2 | —30 | 0.273 | 3.66 | (4) |
| Corkboard | | 12.2 | 0 | 0.282 | 3.55 | (4) |
| Corkboard | | 12.2 | 30 | 0.292 | 3.42 | (4) |
| Corkboard | | 12.2 | 60 | 0.30 | 3.33 | (4) |
| Corkboard | | 12.2 | 90 | 0.309 | 3.24 | (4) |
| Corkboard | | 8.0 | —60 | 0.231 | 4.33 | (4) |
| Corkboard | | 8.0 | —30 | 0.24 | 4.17 | (4) |
| Corkboard | | 8.0 | 0 | 0.249 | 4.01 | (4) |
| Corkboard | | 8.0 | 30 | 0.258 | 3.88 | (4) |
| Corkboard | | 8.0 | 60 | 0.267 | 3.75 | (4) |
| Corkboard | | 8.0 | 90 | 0.276 | 3.62 | (4) |
| Corkboard | | 6.5 | —60 | 0.222 | 4.50 | (4) |
| Corkboard | | 6.5 | —30 | 0.233 | 4.29 | (4) |
| Corkboard | | 6.5 | 0 | 0.244 | 4.10 | (4) |
| Corkboard | | 6.5 | 30 | 0.255 | 3.92 | (4) |
| Corkboard | | 6.5 | 60 | 0.267 | 3.75 | (4) |
| Corkboard | | 6.5 | 90 | 0.276 | 3.62 | (4) |
| Eureka Corkboard | Corkboard with asphalt binder | 14.5 | 90 | 0.32 | 3.13 | (1) |
| Foamglas | Cellular glass slab insulation | 10.3 | —60 | 0.337 | 2.97 | (4) |
| Foamglas | Cellular glass slab insulation | 10.3 | —30 | 0.353 | 2.83 | (4) |
| Foamglas | Cellular glass slab insulation | 10.3 | 0 | 0.371 | 2.70 | (4) |
| Foamglas | Cellular glass slab insulation | 10.3 | 30 | 0.387 | 2.59 | (4) |
| Foamglas | Cellular glass slab insulation | 10.3 | 60 | 0.404 | 2.47 | (4) |
| Foamglas | Cellular glass slab insulation | 10.3 | 90 | 0.422 | 2.37 | (4) |
| Foamglas | Cellular glass slab insulation | 10.5 | 50 | 0.40 | 2.50 | M |
| Foamglas | Cellular glass slab insulation | 10.5 | 300 | 0.55 | 1.82 | M |
| Mineral Wool Board | Made from mineral wool with binder | 15.7 | —60 | 0.253 | 3.96 | (4) |
| Mineral Wool Board | Made from mineral wool with binder | 15.7 | —30 | 0.267 | 3.75 | (4) |
| Mineral Wool Board | Made from mineral wool with binder | 15.7 | 0 | 0.280 | 3.57 | (4) |
| Mineral Wool Board | Made from mineral wool with binder | 15.7 | 30 | 0.294 | 3.40 | (4) |
| Mineral Wool Board | Made from mineral wool with binder | 15.7 | 60 | 0.308 | 3.25 | (4) |
| Mineral | Made from mineral wool with binder | 15.7 | 90 | 0.322 | 3.11 | (4) |
| Natur-Zone | Treated hog hair covered with film of asphalt | 10.0 | 75 | 0.28 | 3.57 | (3) |
| Rock Cork | Rock wool with binder | 16.7 | 90 | 0.37 | 2.70 | (1) |
| Rock Cork | Rock wool with binder | 14.5 | 77 | 0.33 | 3.03 | (1) |
| Shredded Wood and Cement | Slab insulation made from shredded wood and cement | 29.8 | | 0.77 | 1.30 | (4) |
| Thermax | Slab insulation made from shredded wood and cement | 24.2 | 72 | 0.46 | 2.17 | (3) |

References—see last column of tables:

(1) United States Bureau of Standards. (2) Tests conducted at University of Illinois by A. C. Willard, L. A. Harding, L. C. Lichty and others. (3) Tests conducted at Illinois Institute of Technology (Armour Research Foundation), by J. C. Peebles. (4) Tests conducted at University of Minnesota, by F. B. Rowley, et al. (5) American Society of Heating and Ventilating Engineers Research Laboratory. (6) Tests conducted at Pennsylvania State College by Hechler and Queer. (M) Data from manufacturer; authority not stated.

Heating, Piping & Air Conditioning

AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

Journal Section

# Triethylene Glycol Vapor Distribution for Air Sterilization†

By Edward Bigg*, M.D., Chicago, Ill., B. H. Jennings**, and F. C. W. Olson††, Evanston, Ill.

**SUMMARY—Apparatus is described for production and duct distribution of triethylene glycol-vapor for air disinfection in a large building. Uniformly spaced molded ceramic fittings on 9 ft centers delivered glycolized air from the ducts in which the mixture was carried at relatively high static pressure into the upper portion of the rooms with satisfactory results.**

**Individual units for delivering triethylene glycol vapor were used in barracks where experiments were conducted to determine effect of the vapor on incidence of respiratory infection. A reduction of 46.2 per cent in disease incidence was found to result from the use of the triethylene glycol vapor in sleeping quarters.**

THE BASIC concept of air disinfection by the use of glycol vapors has now become well established and repeated laboratory observations have conclusively demonstrated its value (1, 2, 3).[‡] Several published reports on field trials have shown that reduction in airborne infection may be brought about when groups are exposed to bactericidal concentrations of propylene or triethylene glycol vapors (4, 5, 6). It is obvious that further progress will be dependent upon the combined efforts of engineering and medical personnel; the former to develop and refine methods of introduction of vapor, insure effective distribution in the desired space, control desired concentrations, and gather physicochemical data on the behavior of the vapor; the latter to correlate epidemiologic and bacteriologic data on the actual results obtained. The purpose of this paper is to describe additional apparatus and its application and report briefly the results of a controlled experiment at an army training camp.

The original plan of experiment was to treat one large classroom building housing approximately 5000 men; half of these men were to live in glycolized barracks and half in untreated barracks. Trainees in untreated classrooms and barracks were to constitute controls. The sharp reduction in inductees subsequent to the end of the war necessitated changing the program, and consequently it was possible to carry out statistical observations only on the value of glycol vapor in the sleeping quarters (barracks). However, installations had already been made in the classroom building and it was considered worth while to operate this equipment to determine its effectiveness and the feasibility of treating a building of this size, since no attempt had ever been made on such a scale.

## Description of Equipment

### *Building*

This school building was 360 ft wide and 600 ft long, having a capacity of 3,500,000 cu ft, exclud-

†This work was done under contract between Northwestern University and the Office of Scientific Research and Development and the Office of the Surgeon General, U. S. Army.

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‡Numerals in parentheses refer to Bibliography.

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ing the airplane hangar. The roof was of sawtooth construction. Fig. 1 shows a diagrammatic representation of this structure. The interior was divided by metal partitions into 105 rooms. A few of these partitions were completed, extending from the floor to the roof; most were partial, extending from the floor to a point about fourteen feet above the floor. The building was heated by steam; some of the smaller rooms had steam radiators, but most of the rooms depended upon unit heaters. It was obvious that in a building of this type a duct system would be necessary to obtain uniform distribution of glycol throughout the building. The duct layout is also shown in Fig. 1. As may be seen, seven lengths of duct were installed, each to treat a portion of the building, and each operating as an individual unit. Such a unit consisted of the duct, a vaporizer and a high pressure blower fan. The duct handled only recirculated air. Ventilation was obtained by opening windows and roof vents.

*Ducts*

The ducts were constructed of three foot lengths of laminated asbestos. The sections were joined with asbestos collars cemented to the ducts. Uniformly spaced (nine feet) venturi-type openings for distribution were located on the lateral surfaces. Holes were cut through the metal partitions and the ducts were hung by cables attached to sheet metal collars and connected to overhanging guide wires. Each duct was sealed at the far end and since the ducts were from 300 to 460 ft in length, it was desirable to use six inch diameter ducts in order to reduce the pressure drop throughout the system. A total of 2,300 ft of duct was installed.

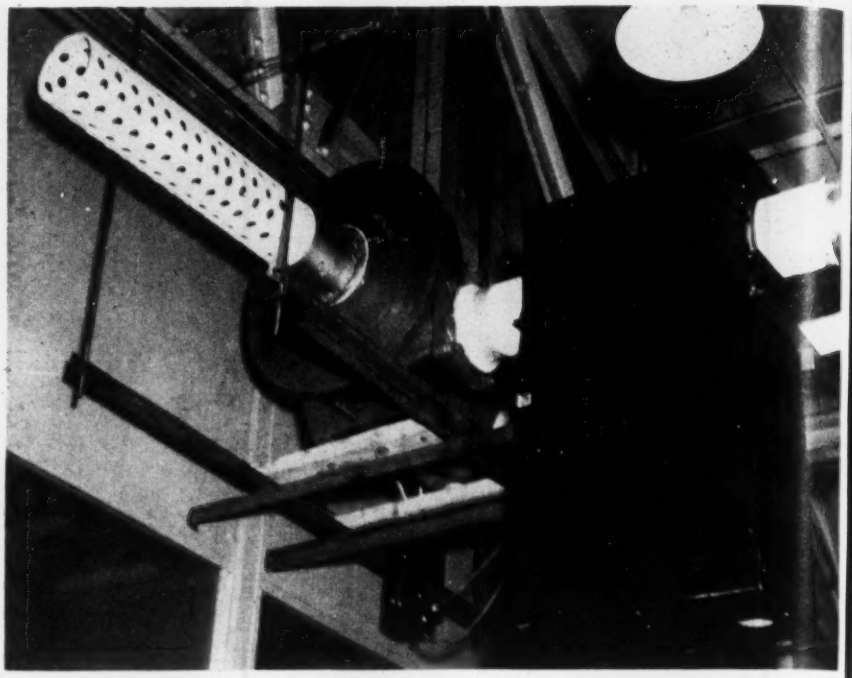


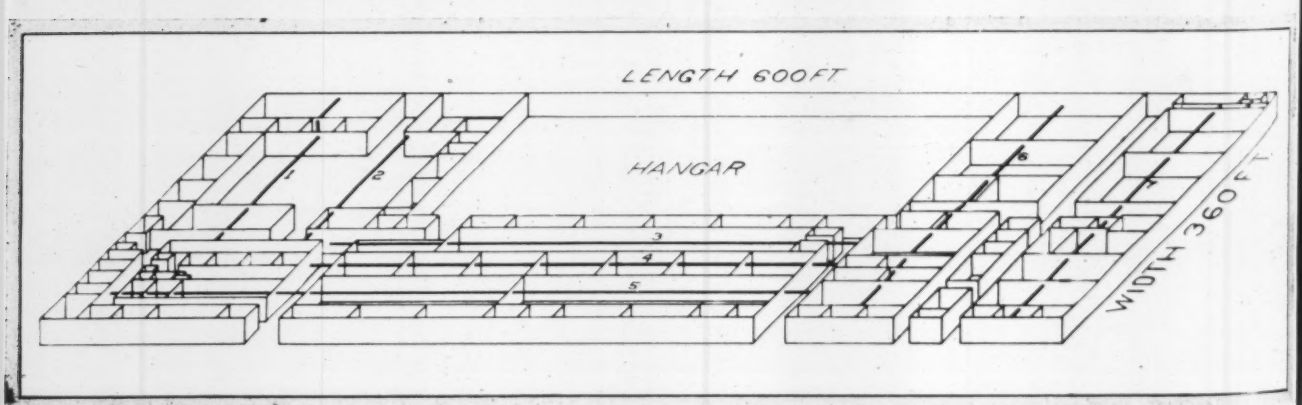
**Fig. 2—Blower fan and glycol vaporizer connected to duct with inlet silencer in place**

*Fans*

The blowers used were of a radial-blade type delivering 800 cfm of air against eight inches of water pressure. The blower was driven by a three horsepower, 220 volt, three phase motor running at 3000 rpm. This high speed caused considerable noise, which was greatly reduced by the installation of a muffler in the air intake. This muffler, shown in Fig. 2, consisted of a three foot length of six inch diameter duct. Forty-eight one inch holes were cut in the side of the duct, a heavy screen was fixed inside the duct six inches from the fan, and steel wool was packed loosely in the duct. This noise reduction method was sufficient in all cases where the blowers were installed in corridors. One blower was installed in a classroom and required a special soundproof box. This box was built of sheet metal and lined with one inch hair felt. Excellent silencing was obtained by this method.

The choice of these blowers was dictated by availability of equipment during the war. It is quite probable that a specially designed blower driven by a one horsepower motor or less would be sufficient. However, it was necessary to use these oversize blowers because they and the motors were readily available and could be fitted to the duct system without special adapters.

**Fig. 1—Diagrammatic representation of glycol treated school building showing seven duct runs**



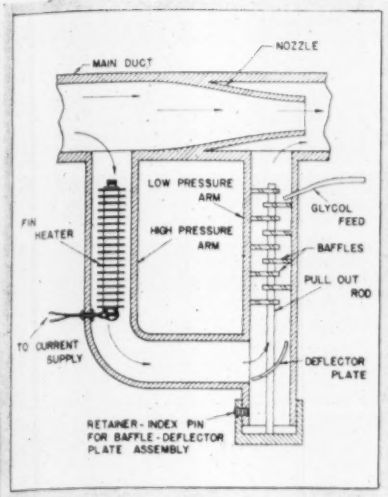



**Fig. 3—Diagrammatic representation of vaporizer aspirator used for glycol generation**

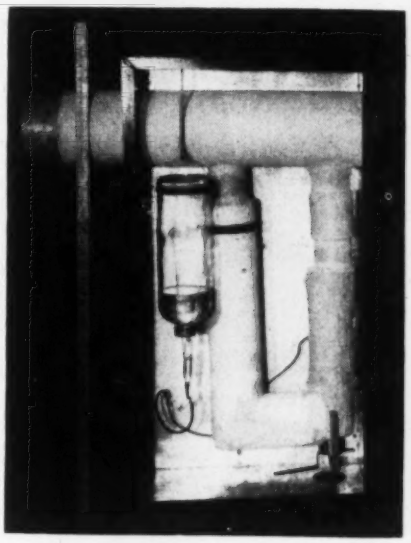


**Fig. 4—Photograph of vaporizer aspirator for glycol generation**

*Vaporizers*

The problem of generating sufficient quantities of vapor and introducing it into the comparatively high pressure air in the ducts was solved by the construction of a *vaporizer-aspirator* shown diagrammatically in Fig. 3 and by photograph in Fig. 4. A constriction was placed in the main duct close to the blower. On the high pressure side, a take off arm permitted air to pass over a strip heater. The heated air then struck a deflector plate and passed upward over a series of baffles. A glycol-water mixture was fed on an upper baffle and then dripped down on successive baffles. The hot air was therefore in contact with a large wetted glycol surface on which vaporization took place. The air-vapor mixture entered the distributing duct in the low pressure area on the other side of the constriction. Insulation was required around the entire unit.

To permit inspection and cleaning of the baffles (which, however, proved to be unnecessary), they were mounted on a pull out rod so that the entire assembly could be withdrawn from the lower part of the unit. Feeding of glycol liquid through a needle valve was satisfactory, although it did require periodic inspection and adjustment.

When heating pure glycol, a fire hazard is present (7). This can be practically eliminated by diluting the glycol with 5 to 10 per cent water which was done. The fin heater was so connected that in the event of blower failure the heater was automatically disconnected.

The vaporizer described was so devised that the quantity of glycol in it at any time was relatively small and remained in the vaporizer a comparatively short time. Therefore, the problem of deterioration of glycol on prolonged heating was practically eliminated. The output of the vaporizer could be closely controlled by varying the feed rate.

*Operation of the System*

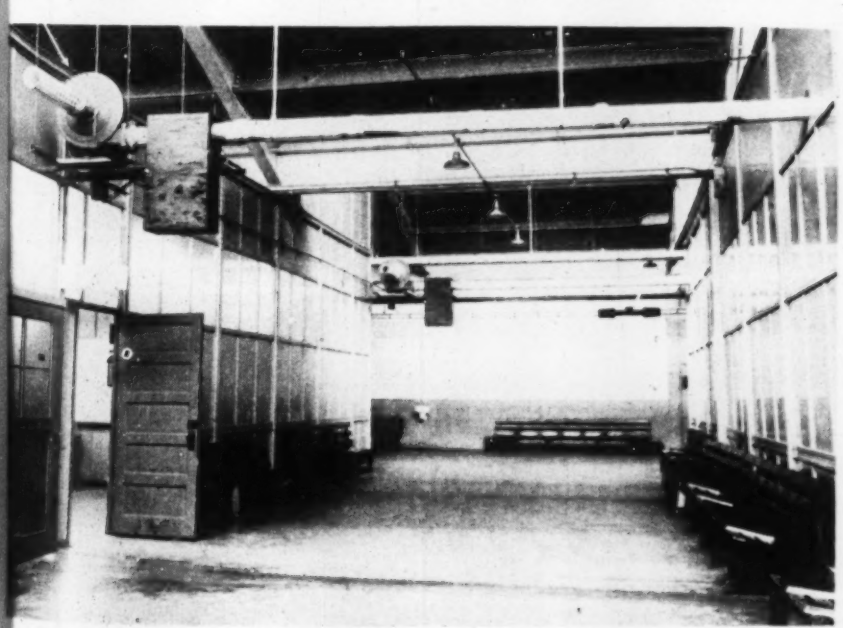
In Fig. 5 is shown the fan with its inlet silencer drawing air from the room and delivering it into the duct system. Close to the start of the duct system can be seen the housing of the high capacity glycol vaporizer, from which glycolized air passed into the duct system, where openings at appropriate intervals controlled the distribution into treated spaces.

Fig. 6 shows a duct in a classroom. The outlets are small molded ceramic fittings placed on approximately nine foot centers, and these delivered the glycol vapor into the upper portions of the room from which the glycolized air was distributed relatively uniformly to all portions of the room.

The concept of the use of a long duct with uniformly spaced small openings for the distribution of air and vapor at a relatively high static pressure is new. It was at first feared by some that most of the air would leave the first few openings and that at the far end of the duct the output would be small, and it was thought that a damper might be required for each opening to regulate the air flow. When the system was finally installed and operating, determinations were made to demonstrate the uniformity of air flow and showed that the output from each opening was practically constant. Table 1 indicates the performance of the ducts at static pressures from 1.7 in. water to 8.0 in.

The glycol vaporizers used in the school building functioned satisfactorily for the purposes of the study,

**Fig. 5—Glycol vapor duct installed with fans and vaporizers**



but certain modifications will be required before such equipment is adaptable for routine application. These would include a positive feed mechanism, alteration in type of baffles, provision for rapidly adjusting and measuring air flow, and increasing the heat input so that a greater quantity of glycol may be vaporized when required. The blower used was larger than required; a smaller blower would reduce the noise substantially and with a lesser static pressure would reduce possibility for droplet carryover.

**Table 1—Output of Duct System**

| Duct No. | Static Pressure at entrance In. $H_2O$ | Ratio of output of last opening to output of first opening |
|---|---|---|
| 1 | 7.0 | 0.96 |
| 2 | 8.0 | 0.94 |
| 3 | 1.7 | 1.00 |
| 4 | 1.8 | 0.94 |

Glycol concentrations were determined daily when the equipment was in operation. With a glycol output per unit of approximately 80 cc per hour, the concentration was maintained between 3 and 5 micrograms triethylene glycol per liter of air in the space treated by that unit. There was a noticeable variation in the concentration from day to day which is best explained by the effect of weather changes and the type of building construction. The openings in the duct were so placed that the glycol vapor was emitted in a horizontal stream. If the roof temperature and other conditions were such as to create an updraft, some of the glycol carried first to the ceiling and escaped through the roof windows before it could be recirculated towards the floor. On the other hand, with a prevailing downdraft, the glycol concentration quickly built up at the floor level. This variation could probably be reduced by placing the duct openings to blow downward at an angle of approximately 45 degrees.

*Treatment of Barracks*

It has been mentioned before that a duct distribution system was employed exclusively in the large school building. Because of the cost factor and other problems involved, it was decided to use individual glycol units in the barracks where the men resided. Because of the necessity of controlling a large number of individual units, these were not as satisfactory as the central system, but did deliver glycol into these spaces. The models available at that time have been considerably improved and modified, but an idea of their arrangement and operation should be of interest.

**Fig. 6—Large classroom showing overhead glycol vapor distribution duct**

Fig. 7 shows one of the units mounted on a post in a barracks. These were operated by electrical heating of a glycol water solution, and a glycol water vapor passed into the barracks space. The vapor can be seen emerging from the outlets of the unit, and this visible vapor very quickly diffused into the atmosphere of the room and is not noticeable. These units were filled manually every two or three days. A more detailed description of the units is given in a previously published paper (8). The control of glycol concentration in the barracks was necessarily crude. The units had three heat inputs which could be varied to give large changes in concentration. Adjustment between these ranges was made by altering the glycol concentration in the vaporizer feed bottle in the manner described (8). Admittedly, this type of control is more an art than a science, but as found by experience, the average service man learned it well enough in a week or two to be able to maintain satisfactory concentrations in the barracks.

**Observations on Disease Incidence**

This study was limited to two squadrons, one of which was exposed to glycolized air in sleeping quarters (barracks). The mean strength of each squadron was approximately 500 men. Every effort was expended to obtain complete comparability of test and control groups since this essentially is the crux of such an experiment. Students were assigned to squadrons for purposes of housing and administration only, and were indiscriminately intermingled in classrooms, mess halls, and recreational activities. Each squadron, within the limits of chance, was composed of personnel with equal distribution as to age, length of military service, crowding in barracks, population turnover, and geographical origin.

Clinical data were based on cases hospitalized for those diseases believed to be disseminated by the aerial route and consisted of the diagnoses placed in the hospital record of patients at the time of discharge, at which time the diagnosis was definitely established. Records included measles, mumps, *common cold*, atypical (virus) pneumonia, bacterial pneumonia, streptococcal infections of the respiratory tract, rheumatic fever, scarlet fever, etc. All cases of respiratory infections of bacterial

nature and those of unknown etiology such as the common respiratory diseases hospitalized less than 10 days after arrival to the camp were omitted. This step was taken to eliminate those cases in which the incubation period is thought to be at least ten days and those in which an infection might have been contracted prior to coming under test observation. Similarly, the specific virus infections, measles, mumps and atypical pneumonia, were excluded in those cases where hospitalization occurred within 21 days. These periods are arbitrary but were applied equally to all men under study, both in test and control group. Disease rates were computed on the basis of rate per 1000 per annum.

A pretest observation period was begun December 1, 1945, and continued for eight weeks. The test period extended for 10 weeks, beginning January 25, 1946, and ending April 5, 1946. During the pretest period a disease rate of 262 per 1000 per annum occurred in the control group and 272 per 1000 per annum in the test group; during the period of *glycolization*, these rates were 714 and 384, respectively. This represents a 46.2 per cent reduction in disease incidence brought about by the introduction of the glycol vapor into the barracks of the test group.

The medical and epidemiologic aspects of the foregoing statistics are to be elaborated and discussed in detail in a forthcoming publication.

## Conclusions

1. The use of an independent duct system carrying glycol vapor and sufficient air for dilution appears the most effective method for distributing glycol vapor throughout large spaces.

2. For small spaces, unit type vaporizers in adequate number and properly located can maintain desired bactericidal concentrations.

3. Hospitalization due to respiratory infection was significantly reduced in groups sleeping in glycol treated quarters confirming previous results of the authors.

## Acknowledgments

The help and advice of E. DeCamp of the Philip Carey Co. in the design and construction of the duct systems is greatly appreciated. In the conduct of the tests and supervision of the military and civilian laboratory personnel the assistance of Major Fred C. Garlock, Sn. C., was invaluable. This investigation was made possible through the whole-hearted cooperation of the Commanding Officers of the Army Air Forces, the staff at Chanute Field and the sponsorship of the Office of the Surgeon General, U. S. Army.

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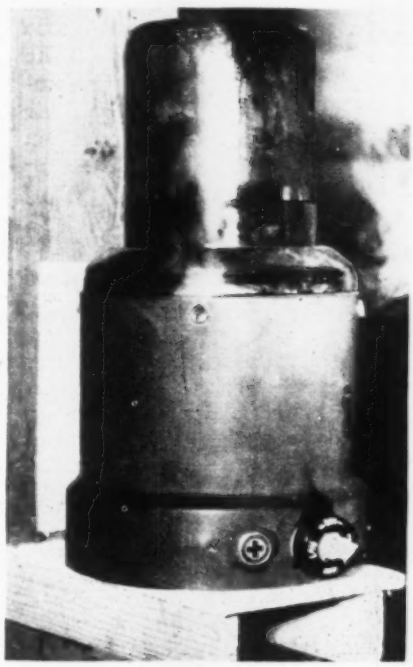
**Fig. 7—Individual type vaporizer used where central system was not installed**





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## PERSONALITY TOPS SCHOLARSHIP IN ENGINEERING SURVEY

According to a recent report issued by the Engineers Joint Council through its committee for Survey of Employer Practice Regarding Engineering Graduates, an employer engages a young engineer and places importance on his personality, not so much on his scholastic record or on engineering experience.

A questionnaire sent to 174 employers of engineers resulted in replies from 104, representing colleges and 19 fields of industry, including instrument, electrical, petroleum, chemical, aircraft, machinery, building, paper, glass, utilities, metal, coal, textiles, soap, rubber, automotive, transportation, shipbuilding and food. In a total of 2,000,000 employed persons, covered by replies to this questionnaire, 40,000 were engineers.

A detailed report based on these replies was published in the April Journals of the Societies making up the Engineers Joint Council.

E. G. Bailey, chairman of the survey committee, stated that the report was prepared in the belief that it would be helpful to junior engineers, employers of engineering graduates, colleges and engineering societies. He expressed his opinion that this was only a preliminary report and that a much wider survey has been recommended to begin next Fall.

Practically all of the 104 surveyed, according to the report, encouraged membership in engineering societies and reported that employees' traveling expenses were paid by employers in order to encourage participation in programs of these societies.

# Proposed Design Procedure for Large Mechanical Warm Air Heating Systems

By S. Konzo*, R. J. Martin**, D. S. Levinson†, and R. W. Roose‡, Urbana, Ill.

**SUMMARY — An exploratory outline of a proposed design procedure for mechanical warm air systems in which the hourly design heat loss exceeds 150,000 Btu. Suggestions are given for determining suitable design values for bonnet and register temperatures, air volume at registers, furnace temperature rise, register pressure loss, and static pressure.**

IN 1940 the Installation Codes Committee of the *National Warm Air Heating and Air Conditioning Association* published a Design Manual No. 7 entitled *Code and Manual for Design and Installation of Warm Air Winter Air Conditioning Systems.* This Manual was intended for smaller structures having design heat losses less than 150,000 Btu per hour and for furnace-blower combinations having the following performance characteristics:

1. Temperature rise through furnace not to exceed 100 F.
2. Static pressure available for overcoming pressure losses in distribution system of 0.20 in. water gage.

A demand was expressed by industry for a similar design manual applicable to structures having a design heat loss greater than 150,000 Btu per hour, and in which a wide range of values for temperature rise, static pressures, and other variables were to be offered in place of the fixed values given in *NWAHACA* Manual No. 7. This study should be considered as an exploratory outline of possible design procedure and should indicate desired areas of additional research. It is hoped that suggestions will be offered by industry which will aid the Codes Committee in formulating a workable procedure that is sufficiently simple for the heating engineers to handle.

In any overall survey of this nature, no single source of reliable reference material is available. The work has consisted primarily of a systematic and thorough search of data from every available source. In this connection liberal use has been made of data from ASHVE publications and particularly those issued under the auspices of the Research Technical Advisory Committee on Air Distribution and Air Friction and the Committee on Code for Testing and Rating Heavy-Duty Furnaces.

## Preliminary Procedure

In general terms the proposed design procedure sets forth the following preliminary steps.

1. Calculation of design heat losses from individual spaces in the structure. This calculation is made by commonly accepted methods, such as stated in the HEATING, VENTILATING, AIR CONDITIONING GUIDE.
2. Drawing of floor plans of structure showing proposed location of registers and return intakes. The type of register, whether wall or ceiling type, and the desired throws and deflections, are to be specified.
3. Laying out a proposed duct system for both the warm air and return air sides of the system. This includes details of types of fittings in the trunk line and in each branch line, and the actual and equivalent lengths of each branch line from bonnet to register. This does not, however, include sizes of ducts or registers which are still to be determined.

## Determination of Air Volume at Register

The volume of heated air required at a warm air register to offset the heat loss from a space is given by an equation of the form:

$$Q = \frac{H}{(60)\,(0.242)\,(d)\,(t_r - t_o)} \quad (1)$$

in which

$Q$ = quantity of air, at density corresponding to register air temperature, in cubic feet per minute.
$H$ = hourly heat loss of space at design condition, including transmission and air infiltration losses, in Btu per hour.
$d$ = density of air corresponding to register temperature, pounds per cubic foot.
$t_r$ = register air temperature, Fahrenheit.
$t_o$ = temperature at return intake, Fahrenheit.

For a given register air temperature, $t_r$, and for an assumed value of 65 F for the return temperature $t_o$, Equation (1) can be expressed in the form:

$$Q = H \times \text{Factor} \quad (2)$$

in which the factor is a function of the register air temperature alone. For example, for a register air temperature of 140 F, the factor is

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‡Special Research Assistant, University of Illinois. Junior Member of ASHVE.

Presented at the 53rd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Joint Session with the *National Warm Air Heating and Air Conditioning Association*, Cleveland, Ohio, January 1947.

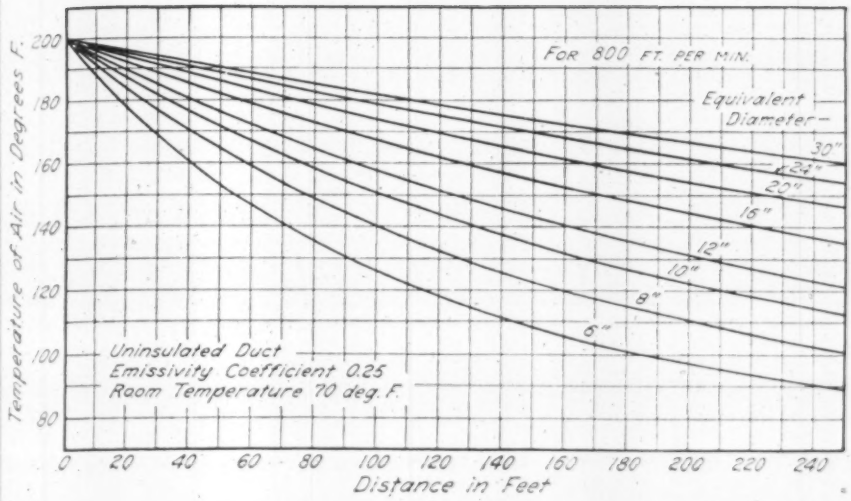


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**Fig. 1—Temperature drop in ducts for air velocity of 800 fpm**

0.014 for 1 Btu per hour, or 14 for 1,000 Btu per hour loss. Therefore, if the register air temperature can be determined, the air volume requirement can then be established for any space for which the hourly heat loss is known.

### Selection of Bonnet Design Temperatures

The temperature of the air leaving any register is dependent upon a large number of variables including temperature of air at entrance to duct, size and length of duct, velocity of air stream, and temperature of ambient air and radiating surfaces. For the purposes of this design manual, data submitted by Kratz, Konzo, and Engdahl[1] as shown in Fig. 1 were utilized. Over a range of bonnet air temperatures from 110 F to 200 F the approximate register air temperatures for varying lengths of ducts, and the corresponding cfm requirements per 1000 Btu per hour loss were determined, as shown in Fig. 2.

In deriving the values in Fig. 2, the assumption was made that the average size of duct was 10 in. in diameter and that the air velocity was 800 fpm. The register temperatures, shown in the upper part of each square in Fig. 2, would be slightly high for velocities less than 800 fpm and for ducts smaller than 10 in. in diameter, and hence would give lower values of cfm requirements for 1000 Btu loss than if corrected values were used. However, calculations made for a 6 in. duct carrying air at 400 fpm velocity indicated that the maximum discrepancy was only of the order of 6 per cent. Conversely, the cfm requirements shown in Fig. 2 would be slightly on the safe side for ducts larger than 10 in. in diameter and for air velocities greater than 800 fpm. Hence, in spite of the fact that the values in Fig. 2 were based upon a specific diameter and velocity, the results obtained from their use should be well within the accuracy required for ordinary calculations.

The Committee on ASHVE Code for Testing and Rating Heavy-Duty Furnaces has under consideration the statement of capacity ratings of the furnace for three or more specific rises of temperature from blower inlet to bonnet of the furnace. It is possible for the installer of such equipment to arbitrarily select one of the temperature rises, and therefore to specify the bonnet temperature to be used for design purposes.

However, for those installations in which a bonnet temperature is not specified, it is proposed that Fig. 2 be used for the selection of a design bonnet temperature. If under normal conditions the maximum and minimum register air temperatures for a given installation are specified as approximately 150 F and 110 F, respectively, it may be noted from the broken lines in Fig. 2 that the selection of a design bonnet temperature can be made as follows:

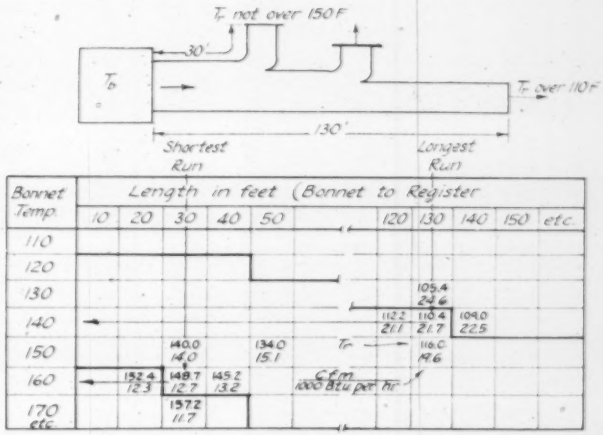
1. Using the longest actual length from bonnet to register, follow the vertical column downwards until the broken line is crossed. Read the maximum permissible temperature for bonnet air in the left-hand column. For example, for an actual length of 130 ft the corresponding maximum bonnet temperature is 140 F.

2. Using the shortest actual length, from bonnet to register, in the same installation, follow the column downwards until the lower broken line is reached. Read the minimum design bonnet temperature in the left-hand column. For example, for an actual length of 30 ft the corresponding minimum bonnet temperature is 160 F.

3. Select any design temperature within the range thus determined. For the example given, a mean temperature of 150 F could be specified, although either 160 F or 140 F could also be used.

4. Regardless of whether the design bonnet temperature is determined arbitrarily or by the preceding method, once selected it must be used

**Fig. 2—Proposed table showing relationships of bonnet temperature, register temperature, and cfm requirements**




Design bonnet temp. can be 140 F, 150 F, or 160 F. Use 150 F.

[1] "Temperature Drop in Ducts for Forced-Air Heating Systems, by A. P. Kratz, S. Konzo, and R. B. Engdahl. (University of Illinois Engineering Experiment Station, *Bulletin* 351, 1944.)

in all later calculations. The cfm requirements per 1000 Btu per hour loss can then be readily determined from the lower of the two numbers in each square in Fig. 2. For example, in the installation under consideration a design bonnet temperature of 150 F can be assumed. For actual lengths from bonnet to register of 30, 50, and 130 ft the corresponding cfm requirements are 14.0, 15.1, and 19.6 respectively, for each 1000 Btu per hour heat loss. The procedure given permits the designer a wide latitude in the selection of a bonnet temperature. At the same time some consideration is given to the heat loss from ducts so that long branch runs are supplied with relatively larger quantities of heated air than are short runs.

### Selection of Blower and Furnace

At this point in the design procedure the following information is available, which will be of assistance in the selection of the blower and furnace:

1. *Cfm air requirements at each register.* The air volume requirements at each register are separately determined by the use of Equation (2). The volumes thus obtained are for an air density corresponding to the temperature at each register, and can be corrected to conditions of standard density of air of 0.075 lb per cubic foot. A summation of these corrected volumes can be considered as the air delivery requirement of the blower, for which the ratings are usually expressed in terms of standard density.

A simpler approach would be to make a summation of the actual cfm air requirements, as obtained from Equation (2), and to arbitrarily assume that this sum is the volume at standard density to be handled by the blower. Over a range of average register air temperatures of from 110 F to 150 F, this procedure will give cfm totals about 8 to 15 per cent greater than those determined by the exact method. However, in view of the fact that some amount of air leakage will occur in the duct system between the blower and the registers, the use of the larger cfm values could be justified, and is therefore recommended.

2. *Temperature rise through the furnace.* With the aid of Fig. 2, the design value of the bonnet temperature was previously determined. The difference in temperature of the air at the bonnet and that entering the blower represents the temperature rise through the unit. In the case of heavy-duty furnaces, ratings based on three specific temperature rises of 50, 75, and 100 F degrees are under consideration. From the standpoint of the heating contractor, rating curves covering a wide range of temperature rises would be more convenient to use, and would permit a closer selection of blower capacity.

3. *Heat delivery at registers, and bonnet capacity of furnace.* The summation of the hourly Btu losses from the rooms or spaces in the structure to be heated is equal in magnitude to the heat delivery at the registers. Furthermore, the required bonnet capacity can be determined from the air volume requirement, as determined from the preceding step, and the temperature rise through the furnace.

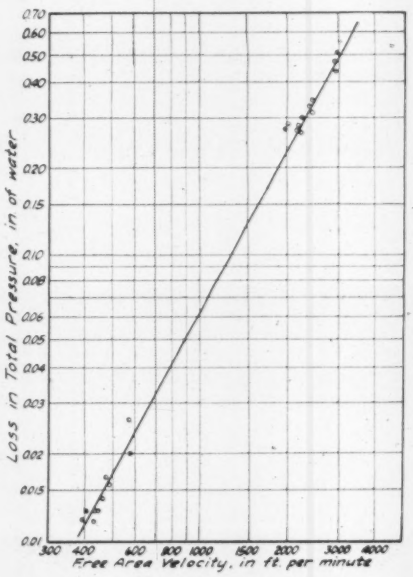
In the case of furnaces to be used in smaller structures (*NWAHACA* Manual No. 7), an allowance of 15 per cent was made to take care of the heat loss of the duct between the bonnet and the registers. That is, when the register delivery was obtained, the required bonnet capacity was determined by dividing the Btu delivery by 0.85. However, in the case of heavy-duty furnace installations, the average length of ducts will vary considerably from one installation to the next, with the result that the heat loss from the ducts will also vary widely. Hence, it is not feasible to state a fixed relationship between register delivery and bonnet capacity.

The location of the furnace with respect to the space to be heated will determine whether the required bonnet capacity or the required register delivery is the significant value to be used in the selection of a furnace. For those installations in which the furnace is used as a *unit heater* and is located in the space to be heated, any duct heat loss may be considered as useful heat and the required bonnet capacity could be considered as equal in magnitude to the required register delivery. However, for those installations in which the furnace is located in a separate heater room only a portion of the duct loss will be regained, and the bonnet capacity alone is considered as the significant item to be used in selecting the furnace. In general, the difference between the bonnet capacity and the register delivery represents the *piping loss* of the system, and may also be considered as representing some amount of reserve capacity.

For those cases in which the equipment is used for buildings to be heated intermittently, some reserve capacity may be needed for the pick-up load. As yet, no decision has been made by the heavy-duty furnace committee as to the desired magnitude of this reserve capacity. Furthermore, no conclusion has been made as to whether the duct system should be designed for the normal load or the pickup load. If the latter procedure is decided upon it may necessitate a revision of the original cfm requirement at each register.

It is obvious that the decisions made by the heavy-duty furnace committee will affect the final procedure for design of duct system and selection of equipment. It is hoped that some uniformity in practice will eventually be obtained.

**Fig. 3—Relationship between free-area velocity of register and total pressure loss**




### Selection of Register Sizes and Pressure Loss

The problem that faces the designer at this point is complex, in that a number of interrelated items must be considered separately and concurrently. Two cases can be presented that illustrate the point.

*Case* 1: A factory assembled heating unit, consisting of a furnace and blower, has a rated bonnet capacity of 300,000 Btu per hour and is equipped with a blower having a rated delivery of 4000 cfm at 0.75 in. static pressure. Assuming that the pressure loss through blower, cabinet, furnace casing, and filters is 0.35 in., the available static pressure for the return and supply duct system is 0.75-0.35, or 0.40 in. If a further assumption is made that the pressure available in the bonnet is 0.20 in., leaving 0.20 in. for the loss in the return side, it is apparent that the total loss of the ducts, fittings, and registers can-

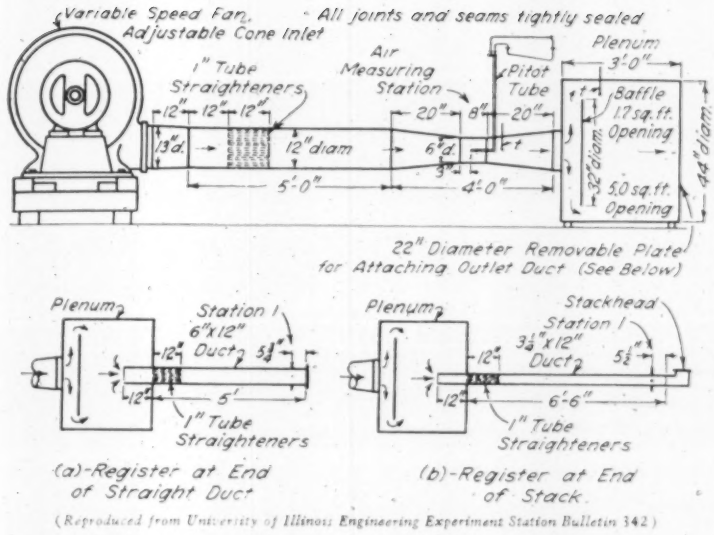



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**Fig. 4—Diagram of Laboratory Plant for measurement of pressure losses in registers and fittings**

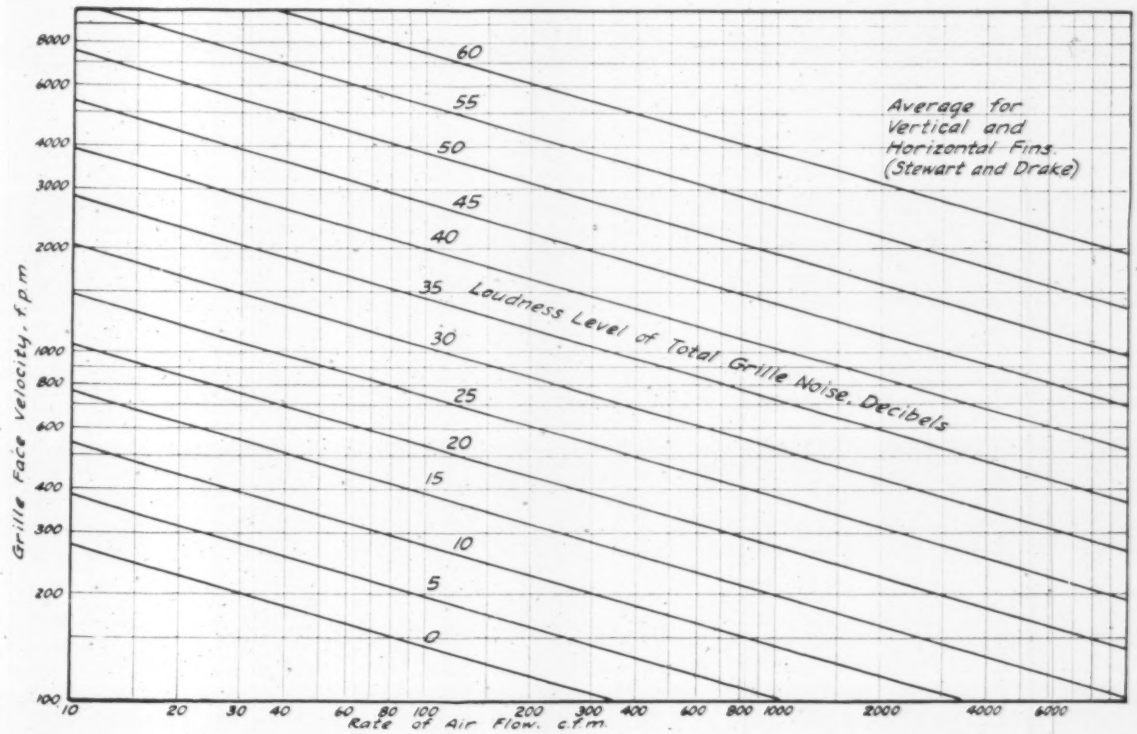
not exceed 0.20 in. For those installations in which relatively long throws of air are required at the register, an available pressure behind the register of 0.20 in. or more would be frequently encountered. In such cases, the 0.20 in. pressure available at the bonnet is not sufficient to take care of both register and duct work. Obviously, a better procedure would be to first determine the register loss and then decide what pressure is available for the ducts.

*Case* 2: A designer has the option of selecting a blower independently of the furnace. For example, a furnace with a rated bonnet capacity of 300,000 Btu per hour can be installed with a blower having a rated delivery of 4000 cfm at 0.75 in., or one having a delivery of 4000 cfm at 1.0 in. The latter blower would probably be larger in size, greater in initial cost, and more expensive to operate than the former. On the other hand a greater allowable pressure would permit the use of high-pressure registers having long throws, as well as smaller-sized duct work. In this case also, before any selection of blower or duct friction loss can be made, some evaluation of pressure loss at the register is necessary as a first step.

A thorough search of a number of references, including practically all manufacturers' catalogues, indicated the need of a summary table embodying complete performance characteristics of a wide range of register sizes. Wherever such data for a specified model are stated in the catalogue, they are preferred to any typical results that may be derived for an *average* register. Nevertheless, the discrepancies that were noted in the data from various sources, as well as partially complete listings in many sources of material, led to the formulation of a table that will be explained and shown later.

This table when completed will show a summary of the best available data from a number of sources, for registers of commercial types having horizontal or vertical bars, and for a zero degree deflection angle of the air stream. The table does not apply to perforated grille type faces. Similar tables are proposed, but not shown, for bar-type registers having angles of deflection of 22 deg and 30 deg.

**Fig. 5—Loudness levels for registers.** ***(Average values for vertical and horizontal fins. Data from Stewart and Drake)***



For a given throw and air volume to be handled, the following information can be obtained: (*a*) Register free areas, square inches; (*b*) Pressure loss, inches of water; (*c*) Sound level, decibels. In addition, information relative to: (*d*) Selection of nominal register size, and (*e*) Effect of deflection angles on pressure loss can also be obtained.

(*a*) *Relation of area to throw:* The data presented by Tuve and Priester[2] were selected as the most comprehensive found in a number of references that were investigated. The basic equation proposed by Tuve and Priester was of the form:

$$V_r = \frac{KQ}{X\sqrt{A_e}} \quad \ldots\ldots\ldots\ldots\ldots\ldots (3)$$

in which

$V_r$ = residual maximum velocity, feet per minute.
$K$ = constant.
$Q$ = volume of air discharged, cubic feet per minute.
$X$ = throw of air stream, feet.
$A_e$ = effective area of outlet, square feet.

Furthermore,

$$A_c = A_e/C \quad \ldots\ldots\ldots\ldots\ldots\ldots (4)$$

and

$$V_c = Q/A_c, \text{ or } A_c = Q/V \quad \ldots\ldots (5)$$

in which

$A_c$ = core area of register, square feet
$C$ = coefficient of discharge
$V_c$ = core velocity, feet per min

[2]Control of Air Streams in Large Spaces, by G. L. Tuve and G. B. Priester. (ASHVE TRANSACTIONS, 1944, p. 153.)

By rearranging Equations (3), (4), and (5), the following form was derived:

$$V_c = \frac{X^2}{Q}\left(\frac{CV_r^2}{K^2}\right) \quad \ldots\ldots (6)$$

Numerical values for $V_r = 50$ fpm, $K = 2.5$, and $C = 0.61$ were inserted in Equation (6) and the relationship was simplified to the form:

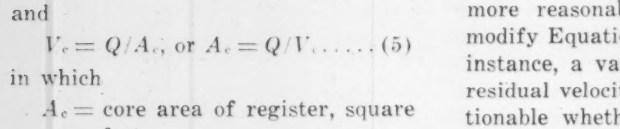
$$V_c = \frac{X^2\,(244)}{Q} \quad \ldots\ldots\ldots (7)$$

By substituting numerical values for $X$ and $Q$ in Equation (7), values of $V_c$ were obtained, and these in turn were substituted in Equation (5) to give the required core areas, $A_c$.

In order to obtain free areas, which are usually stated in manufacturers' catalogues, the assumption was made following consultation with the register manufacturers, that the ratio of free area to core area was 0.78. Hence, by the use of a modified version of the Tuve-Priester equation, (Equation 3), relationships between throw, volume, and free areas of registers could be obtained.

In the opinion of some plant designers who have been consulted, a residual velocity of 50 fpm at the wall opposite a register was considered as too large. In their opinion, average stream velocities of the order of 15 to 20 fpm were considered as more reasonable. It is possible to modify Equation (7) by inserting, for instance, a value of 15 fpm, for the residual velocity. However, it is questionable whether such an extrapolation to velocities which are practically unmeasurable can be justified. The suggestion has been made that the length of *throw* should be considered as equal to say ¾ of the width of the space, instead of the entire width, and that the values obtained from Equation (7) be applied to this shortened distance. Since no definite research results are available at the present time, it is probable that the use of this arbitrary and inadequate rule may be necessary as a temporary expedient.

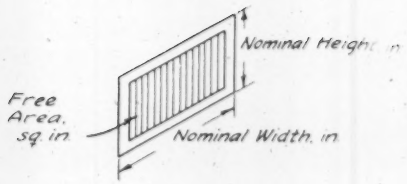
**Fig. 6—Proposed table giving relationships of air throw, free-area, pressure loss, and sound level for registers *(for registers of the non-deflecting type)***




| Quantity, cfm | Throw, ft | | | | | | |
|---|---|---|---|---|---|---|---|
| | 10-12 | 13-15 | 16-18 | 19-21 | 22-24 | 25-29 | etc. |
| 60-99 | 24 / 0.02 | 15 / 0.04 | 10 / 0.08 | 8 / 0.14 | 7 / 0.24 ← Free Area Loss | | |
| 100-139 | 55 / 0.01 | 34 / 0.02 | 23 / 0.04 | 17 / 0.07 | 13 / 0.11 | 9 / 0.21 | |
| 140-179 | | 60 / 0.01 | 41 / 0.02 | 30 / 0.04 | 22 / 0.06 | 16 / 0.12 | B |
| 180-219 | | 95 / 0.01 | 59 / 0.01 | 47 / 0.03 | 35 / 0.04 | 25 / 0.08 | 35 decibel limit |
| 220-259 | | | 92 / 0.01 | 66 / 0.02 | 50 / 0.03 | 36 / 0.06 | |
| etc. | | | etc. | | A | | |
| | | | | | 20 decibel limit | | |

REGISTER THROW, FREE AREA, AND LOSS.

(*b*) *Relation of Velocity to Pressure Loss.* The pressure loss through a register can be expressed in terms of (*a*) inches of water, (*b*) velocity head loss, or (*c*) equivalent feet of duct leading to the register. From the standpoint of the designer, pressure losses stated in terms of equivalent feet of duct are most convenient to use, since resistance of fittings can be similarly expressed. However, in the cases of large installations no definite relationship exists between the size of the register and the size of the duct leading to it. Hence, no feasible method could be devised for expressing pressure losses of registers in terms of equivalent feet of duct. Neither did an expression of pressure losses in terms of velocity heads offer any simplification of approach. Hence the decision was made to express losses for registers in terms of inches of water.




| Free Area, sq in. | Nominal Register Size, in | | | | | | |
|---|---|---|---|---|---|---|---|
| 18 or less | 8-4 | | | | | | |
| 19-26 | 8-5 | 10-4 | | | | | |
| 27-32 | 8-6 | 10-5 | 12-4 | | | | |
| 33-42 | | 10-6 | 12-5 | 14-4 | 16-4 | | |
| 43-52 | | 10-8 | 12-6 | 14-5 | 16-5 | 18-4 | 20-4 |
| etc. | | | etc. | | | | |

**Fig. 7—Proposed register selection table**

For this relationship, data from Kratz and Konzo[3] were used. Fig. 3 shows the relation between the free area velocity and total pressure head of the register for 23 types of commercial models of widely varying design, in which no air deflection was used. These tests were conducted in a plant shown in Fig. 4. The total pressure ahead of the register included not only the losses due to friction, turbulence, and expansion, but also the velocity head of the air leaving the register.

(*c*) *Sound Level:* The work of Stewart and Drake[4] on sound level of registers was selected from a number of references consulted. A communication from D. J. Stewart recommended that in place of the figure now in current use,[5] which is for hor-

[3]Pressure Losses in Registers and Stackheads in Forced Warm Air Heating, by A. P. Kratz and S. Konzo. (University of Illinois, Engineering Experiment Station *Bulletin*. 342, 1942. Data shown in Fig. 4 were plotted from data in Table 2, page 115, from the same reference.)

[4]The Noise Characteristics of Air Supply Outlets, by D. J. Stewart and G. F. Drake. (ASHVE TRANSACTIONS, Vol. 43, 1937, p. 81.)

[5]Air Flow and Loudness Chart. HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1946, p. 773, Fig. 5.

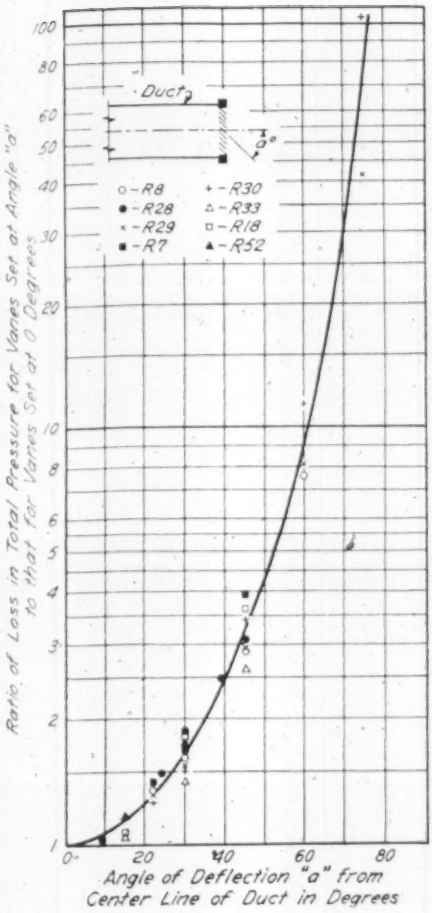



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**Fig. 8—Increase in pressure loss with different angles of deflection**

izontal fin type registers only, that a new curve be drawn based on averages of results obtained by the same authors, for registers with horizontal and vertical fins.[6] The curves thus obtained are shown in Fig. 5, and values read from the curves were used to locate the diagonal broken lines in Fig. 6.

(*d*) *Relation of Free Area to Nominal Register Size:* It is realized that some differences exist in the ratio of free area to the nominal area of registers, especially in the case of perforated grille types. However, a survey of a number of manufacturer's catalogues indicated that the divergence in the ratio of free area to nominal area was not too great in the case of fin or bar type registers, and certainly not great enough to preclude the establishment of a register selection chart as given in Fig. 7. The nominal sizes of registers indicated in this figure may be subject to some changes. However, they do represent the sizes which a number of manufacturers have indicated as being desirable.

(*e*) *Effect of Deflection Angles on Pressure Loss*: Tables similar to Fig. 6 are proposed for registers having deflection angles of 22 deg and 30 deg. For those portions of the table referring to pressure loss, the relationships[7] shown in Fig. 8 will be used. The data were obtained from eight different registers tested in the plant shown in Fig. 4. It is apparent that the pressure loss increased relatively slowly when the angle of deflection was increased from 0 deg to about 15 deg, but increased at a greatly accelerated rate for angles larger than 15 deg.

In many respects Fig. 6 represents an audacious move to generalize data that are inherently not subject to generalization. The data are offered as practical epedients which may aid the designer; until the time is reached when a majority of manufacturers will be able to offer similar data for their specific models.

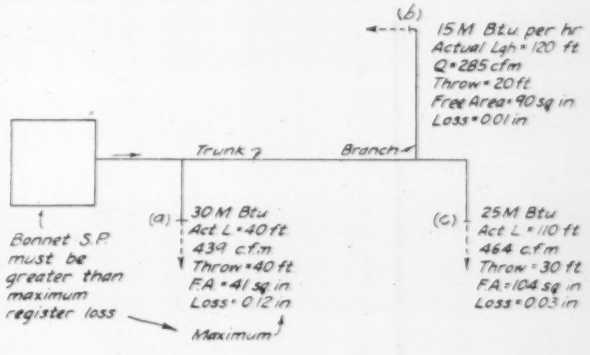
## Selection of Design Static Pressures for Duct System

The pressure loss of each register in the duct system can be obtained by the use of data similar to those shown in Fig. 6. Furthermore, as discussed in a preceding section entitled *Selection of Register Size and Pressure Loss*, the available pressure in the bonnet will be specified or assumed. In any case, it is apparent that the bonnet pressure will have to be greater than the maximum pressure loss of any register in the system. The differences between the bonnet pressure and the pressure loss of each register will represent the available pressure loss for the duct system connecting the bonnet to the register.

For example, as shown in Fig. 9, if the pressure loss of register (*a*) is 0.12 in., for register (*b*) is 0.01 in., and for register (*c*) is 0.03 in., then the maximum pressure loss is 0.12 in. The required bonnet pressure must be greater than 0.12 in. For example, if the bonnet pressure of the unit selected is 0.25 in., then the available pressure for overcoming resistance in each of the runs from the bonnet to the register will be for (*a*) 0.13 in., for (*b*) 0.24 in., and for (*c*) 0.22 in.

Once the total pressure loss for each run has been determined, the selection of the friction loss per 100 ft of duct for each run can be determined by dividing the available pressure for each run by the respective equivalent lengths of ducts from bonnet to register. The design procedure from this point forward is no different from that of a commonly used method, frequently referred to as the *equal friction pressure loss method*. For each branch duct in the system, both the cfm to be handled and the friction loss per 100 ft will be known. By the use of the common friction chart for air flow, the size of the ducts can be readily determined. A few refinements are proposed in the new *NWAHACA* Manual that will simplify the calculation required. For example, a table will be shown in place of the friction chart, and a separate table will be given for the relationship between round pipe sizes and equivalent sizes of rectangular ducts. Furthermore, a complete work sheet, which will enable a designer to follow a step-by-step procedure, is in preparation. Hence, by the use of the prepared tables and work sheets, it is the intent of the Installation Codes Committee to furnish the designer and installer with working tools that will enable them to design a duct system for large, mechanical warm-air systems.

**Fig. 9—Maximum register loss and its relation to duct loss and bonnet static pressure**




[6]Material furnished by D. J. Stewart.

[7]Pressure Losses in Registers and Stackheads in Forced Warm Air Heating, by A. P. Kratz and S. Konzo. (University of Illinois Engineering Experiment Station. *Bulletin* 342, 1942, p. 31.)

In this brief review, it is not possible to show in full detail all the proposed tables, nor to discuss in detail each of the many factors that were considered. Some important omissions have been made, but have not been overlooked. The authors are cognizant of the necessity of taking into account such items as: relation of ventilation load to both cfm requirements at register and furnace capacity; duct design to take care of both cooling load and heating load; magnitude of pressure loss through heavy-duty furnace units; a procedure for the selection of ceiling diffuser type outlets; effect of insulation of ducts on cfm requirements, and many others. In view of the fact that the proposed procedure is still in the draft stage, any suggestions relative to the problem will be gratefully received.

# Discussion

## Friction Charts for Gases Including Correction for Temperature, Viscosity and Pipe Roughness*

**By Richard D. Madison and Walter R. Elliot**

WALTER KUPCHICK, Harvey, Ill.: Your article in the October issue on Friction Charts for Gases Including Correction for Temperature, Viscosity and Pipe Roughness leads me to the following discussion:

Articles on fluid flow have been abundant, and the Fanning equation has been nomographed, charted, and tabulated in a number of different ways. Each author has his special way of expressing the same thing which supposedly makes the computation of fluid flow problems easier for the practicing engineer. However, with the exception of the charts prepared for the common fluids (air, water and steam) these so-called simplifications tend to confuse. It seems absurd to solve a problem in steam flow by applying correction factors to air charts. This is not sound engineering and never will be.

The fundamentals of fluid flow are not so profound that a lot of hocus-pocus in the form of fancy charts is required to clarify the subject for the engineer. If the engineer is concerned with a fluid—other than air, water, or steam he can solve the problem very easily if he knows the properties of the fluid and makes reasonable assumptions in his design.

It is the assumptions that are really critical in a problem, and to assist in making them one finds that there is a great deal of basic data lacking in handbooks, texts, and magazines. For example, in calculating pressure drop in piping for liquids, it is customary to express losses through fittings and valves in terms of equivalent length of pipe. Is it reasonable to assume that these losses are the same regardless of the velocity of the fluid? Probably the variation is not great for the usual range of velocities selected in turbulent flow, but how important are losses in fittings and valves in viscous flow?

Viscous flow is a neglected subject and is usually skipped over lightly in fluid flow articles. Yet, the practicing engineer has occasion to deal with viscous liquids. In pumping problems the horsepower requirements are difficult to estimate. I have heard one pump expert (who is quite familiar with the Fanning formula and Reynolds Number) state that for viscous flow he calculates the horsepower required in the usual way, then multiplies the result by four to get a motor that will do the job. This man has learned by experience. Now, why is such a relatively crude procedure necessary? Surely enough data on viscous flow should be available to calculate within 20 per cent of the actual requirements, a not-too-great refinement for turbulent flow problems.

It is surprising how much trial and error the engineer in the field must resort to because of the lack of basic engineering data. It is time that technical editors begin utilizing the space in their magazines more effectively, cutting out some of these articles on super-refined research that corroborates previous research and articles on the fancy analysis of established formulas. One of the important fields that needs resurveying in fluid flow.

AUTHORS' CLOSURE: Unfortunately, the reader has not grasped some of the significance of this article. The subject of fluid flow is not so simple as Mr. Kupchick would lead one to believe. If so, he would not pose a few questions and then conclude by saying it is a field that needs resurveying. The authors feel that they have found a unique and simple method of expressing flow in the turbulent region, and that it is representative of the latest information on the subject. The authors know the reluctance of many engineers to use equations in connection with the Reynolds Number. The charts presented thus avoid the use of the Reynolds or other dimensionless numbers, but the derivation is given for the engineer who wishes to know the origin of the material he uses rather than "hocus-pocus" data, to which Mr. Kupchick makes reference.

The purpose in giving several examples, is to show how versatile the chart, Fig. 3, really is. We do not recommend it for fluids other than air if more suitable charts for these fluids are available. However, many charts have decided limitations and we are offering this one to cover a broad range, but without loss in basic accuracy. For example, if, upon looking at the chart, Fig. 3, one finds that most of the work lies in a narrow range of velocity and pipe diameter than one can use the value so indicated as a constant without serious error. In this way, the chart will act as an aid in making some of the customary assumptions less critical, and the authors agree that anything to help lessen the errors due to assumptions is to be desired.

Mr. Kupchick doesn't differentiate between viscous liquids and viscous flow. The viscous flow formula is simple and most authors do not pass it over lightly, but there is less to be said about it. The flow of viscous fluids in the turbulent region is more difficult to evaluate, and it is with just this thought in mind that the authors have presented these data so when the viscosity of the fluid is known the solution can be made easier and with less guess work. With the tolerance of *Heating, Piping & Air Conditioning* readers the authors may at some time in the future present a similar chart for liquids.

We agree that many engineers "can solve problems very easily" (but not accurately) by making assumptions on their designs. The "pump expert" who multiplies his calculation by four is introducing a practice that is not well to recommend generally. Increasing the motor horsepower by four does not insure that the pump will be correctly suited to the job.

The authors are well aware that in most flow problems, the resistance of fittings is greater than that due to straight pipe friction. The question that is posed about whether to figure fitting losses upon equivalent length or head loss is well taken. Where the elbow or fitting is of considerable length and has small turbulent loss, the equivalent length method is justifiable. Where the elbow or fitting is short and of high loss, the resistance will vary more nearly as the square of the velocity pressure and the method of presenting the loss in percentage of velocity head is more accurate. Many designers ignore this principle and use the method with which they are more familiar.

*Published in ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning*, October, 1946, p. 107.

# Meteorological Instruments and Heating Design

By W. E. K. Middleton*, Ottawa, Canada

**SUMMARY—The measurement of the meteorological elements of most interest to heating and ventilating engineers is discussed in general terms, with particular reference to the limitations imposed on the accuracy of such measurements by the properties of the atmosphere. The recommended exposure of meteorological instruments is also discussed.**

A KNOWLEDGE of the properties of the outdoor atmosphere lies at the basis of heating and ventilating design. The engineer must know not only the average environment of his structures, but also the extreme atmospheric conditions that they are likely to encounter. It is the business of the climatologist to furnish all such information, on the basis of long-continued observations of the various meteorological elements, and most of these observations require the use of instruments.

To the heating and ventilating engineer some of the meteorolgical elements are of much more importance than others. The height of the clouds, for example, is of little concern to him; nor need he worry about the range of vision. The elements with which he is directly concerned are four in number—temperature, humidity, wind, and radiation. By radiation is meant chiefly that which comes from the sun and sky, though the nocturnal radiation from terrestrial objects into space has some influence on heat loss through roofs and walls at night.

*Optics Section, Division of Physics and Electrical Engineering, National Research Council.

## The Exposure of Meteorological Instruments

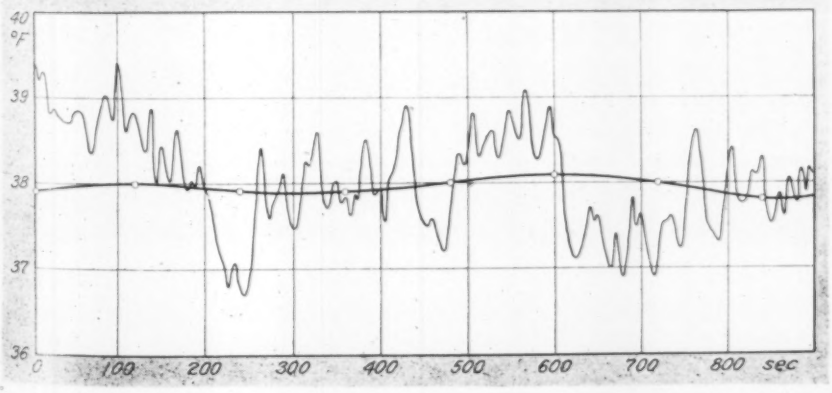
It is a fact not sufficiently appreciated, even by meteorologists, that the properties of the atmosphere itself put a limit to the possible accuracy of meteorological instruments[1]. It is one of the most important characteristics of the lower atmosphere that it is almost always in turbulent motion more or less violent. This turbulence is frequently made evident by the smoke from a chimney; but if the motion of the entire atmosphere on a windy day could be seen, the eddies would probably be far more spectacular than the water eddies of the Niagara gorge.

[1]The Present-Day Accuracy of Meteorological Instruments, by W. E. K. Middleton. (*Quarterly Journal of the Royal Meteorological Society*, 72: 32-50, 1946.)

The effect of this turbulence is to bring samples of air of different properties successively into the vicinity of an instrument. At one instant, the air which reaches the instrument may recently have been in contact with the ground; the next sample may have been brought down from a considerable height; and thus an instrument with a small time constant may give widely fluctuating indications. The effect of these rapid changes on a sensitive thermometer is shown by the wavy line in Fig. 1, the thicker, less irregular line being derived from simultaneous observations of a thermometer having a much greater lag. A knowledge of the magnitude of such effects is of importance to those who design or choose meteorological instruments.

Another aspect of the subject which receives insufficient attention is the wide variation of the properties of the atmosphere from place to nearby place at the same instant. Fig. 2, reproduced from a paper by

**Fig. 1—Effect of instrument sensitivity on indicated temperature during rapid temperature fluctuations. The wavy line shows widely fluctuating temperatures indicated by a sensitive thermometer, whereas the more regular line was obtained from simultaneous observations of a thermometer having a much greater lag**



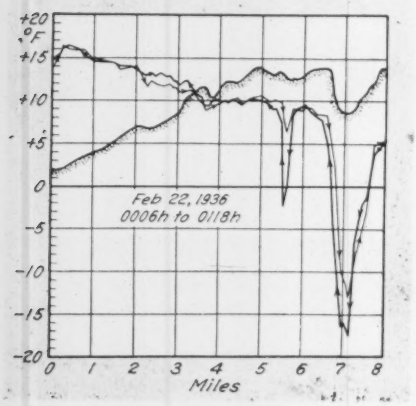
Middleton and Millar[2], shows temperature variations occurring over a distance of 8 miles in the city of Toronto, obtained by attaching a sensitive electrical thermometer to a boom projecting about two feet in front of an automobile and driving as rapidly as traffic permitted from the lake shore inland and back again. The stippled line is an engineer's profile of the street; the maximum difference in elevation is about 300 ft. The occasion was a clear, calm winter night, and afforded a really spectacular example. It will be noted that in the half mile between miles 6.6 and 7.1 there was a temperature difference of about 26 Fahrenheit degrees. In cloudy weather with a strong wind there may be only a degree or two difference over the entire distance. On sunny days the differences are, in general, much less pronounced, the temperatures farther inland being the higher.

It should be emphasized that the calm weather and clear skies which lead to conditions such as those depicted in Fig. 2 tend to suppress the transient variations shown in Fig. 1, so that the two phenomena are entirely distinct. The second is the more important to the heating engineer, because faulty location of observing stations may give an incorrect idea of extreme and mean temperatures in a region. The differences do not vary symmetrically through the twenty four hours, and monthly mean, maximum, and minimum temperatures may not be representative unless the site of a climatological station is chosen with an eye to the topography.

Similar considerations obviously apply to measurements of wind, and less obviously to measurements of humidity. The dew point is, of course, much less affected by local conditions, but non-representative readings may occur in a valley at night. Emphasis has been given to the phenomena discussed because they are fundamental to all meteorological measurements. City meteorology is a somewhat neglected branch of the science[3], although a very large proportion of the taxpayers live in cities. The wind in the central portions of large cities differs greatly in both speed and turbulence from that in the country immediately adjacent, a fact that must be of importance to heating engineers.

[2]Temperature Profiles in Toronto, by W. E. K. Middleton and F. G. Millar. (*Journal of the Royal Astronomical Society of Canada* 30: 265-272, 1936.)

**Fig. 2—Temperature variations found in traveling (both ways) over an 8-mile course in Toronto, Feb. 22 1936, from 12:06 a.m. to 1:18 a.m. The street profile, with a maximum difference in elevation of about 300 ft, is indicated by the stippled line**




## Thermometers

After the phenomena discussed in the preceding section have received due consideration, it remains to discuss the instruments themselves and their exposure. Thermometers are familiar to nearly everybody, and certainly to heating and ventilating engineers. Meteorological thermometers may be indicating or recording instruments; at most climatological stations only indicating thermometers are used. The normal complement is four thermometers: one to register maximum temperatures, one minimum, and two ordinary thermometers, one having its bulb covered with wet cloth to act as a psychrometer in conjunction with the other. At main stations there is usually a thermograph to make a continuous record of the temperature and this is usually a bimetallic or Bourdon tube instrument, or less frequently, an electrical resistance thermometer of the industrial type familiar to engineers.

To obtain real air temperatures the thermometers must be properly exposed. The temperature of any body in the air is a function of the thermal equilibrium between the body and its surroundings. Heat is transferred by three processes, namely conduction, convection, and radiation, and in order that the bulb of a thermometer shall be at the temperature of the air, arrangement must be made to keep the transfer of heat by radiation small in comparison to the transfer by conduction and convection together. This may be accomplished in one or both of two ways: (1) By ventilating the thermometer artificially and so increasing the heat transfer by convection, (2) By greatly reducing the radiation of energy to or from the bulb by surrounding it with surfaces having nearly the temperature of the air. In the Assmann psychrometer (Fig 3) both these means are employed and this is the standard instrument for measuring meteorological temperatures and humidities. At normal climatological stations, the familiar louvered-screen type of instrument shelter is employed, the intention being that the inside of the walls should present to the thermometer a continuous surface at about the temperature of the air, while a fairly free motion of air past the thermometers is maintained. In bright sunshine the illuminated walls will be somewhat warmer than the air, especially when the wind is light.

Few laymen realize the magnitude of the errors that can result from improper exposure. A thermometer exposed to half the sky on a north wall may read 10 Fahrenheit degrees lower than the temperature within the shelter if the night is very clear.

## Hygrometers

The measurement of humidity is an immense subject, the treatment of which would require a large book. This paper will, however, be limited to the mention of two instruments: (1) the wet and dry bulb psychrometer, (2) the hair hygrometer.

A word on nomenclature may not be amiss. *Hygrometer* is a generic term for all instruments which measure humidity. In accordance with international usage, the word *psychrometer* (an unfortunate word that really signifies *cold-meter*) should be reserved exclusively for

[3]There is at least one good book on the subject: Kratzer, A., *Das Stadtklima*, Braunschweig 1937, Vieweg and Sohn.

the wet and dry bulb instrument, if used at all. Its use perpetuates the bad judgment of August, who named it in 1825. The term is superfluous and could be dropped without serious inconvenience.

The wet and dry bulb psychrometer is certainly familiar to all engineers, and it may be that this leads some of us to over estimate the accuracy of the instrument. At ordinary indoor temperatures the psychrometer is amply accurate enough for all technical purposes provided the following conditions are met:

1. The thermometers used have been calibrated and the proper corrections to be applied are known.
2. The wet bulb is covered with thin material which has been carefully washed, and finally rinsed in distilled water.
3. The wet bulb covering is moistened with *distilled* water.
4. The bulbs are ventilated at an air speed of not less than 10 fps.

The last of these conditions is the one most frequently neglected. It can conveniently be met by means of the sling psychrometer, but it should be noted that such an instrument, if of ordinary dimensions, must be rotated about 3 times per second in order to obtain an adequate rate of ventilation.

The use of the psychrometer out of doors requires greater care, especially in winter. At temperatures between 32 F and about +10 F the covering of the wet bulb should be retained, but if a wick is employed it should be removed, the bulb being moistened by dipping it into water at about freezing point. At still lower temperatures it is better to form a layer of ice directly on the bare bulb. In either case it is important that the bulb should be covered with ice and not with super-cooled water. With a textile covering on the bulb, this can remain undetected by observation of the bulb itself, but the proper freezing of the water is easily ascertained by watching the indications of the thermometer as it cools down. As the water freezes, the temperature will rise rather suddenly to 32 F and remain there for a short time before falling to its final temperature. If it does not behave in this way, but falls steadily, it is probable that the water has remained unfrozen.

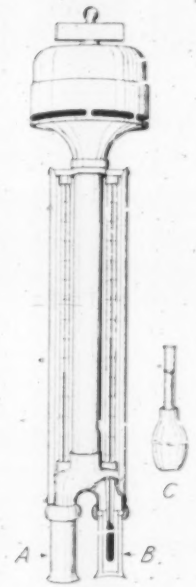
Under some winter conditions the ice covered wet bulb may read a few tenths of a degree higher than the dry bulb. This occurs fairly frequently in moist winter climates, and indicates supersaturation with respect to ice. The usual tables do not cover this range, and the relative humidity is generally recorded as 100 per cent when this phenomenon occurs.

Special attention should be paid to the calibration of thermometers to be used for measuring humidity in winter, since small errors in temperature become important.

The same precautions should be used when the humidity is to be measured in cold storage rooms. At temperatures below 0 F the method, in common with most other methods of measuring humidity, becomes much less accurate.

The hair hygrometer is extensively used for *humidity control*, less often for the *measurement of humidity*. It is in no sense an absolute instrument, and requires frequent comparison with some other kind of hygrometer. One of its most striking (and least understood) features is that it always indicates relative humidity with respect to saturation over water, even at temperatures far below freezing. In comparing its readings with those of the psychrometer under such conditions, it is necessary to know whether the psychrometer tables refer to saturation over water or over ice; if the latter (as is usual), the readings of the hair hygrometer must be multiplied by the ratio of the vapor pressure over ice to that over water at the current air temperature.

Fig. 3—Assmann psychrometer




There are several other methods of measuring humidity, but none has found use in general meteorological practice.

For measurements out of doors, hygrometers (except the sling psychrometer and similar instruments) will normally be exposed in a louvered shelter.

### Anemometers

Every heating engineer knows that the speed of the wind past a wall has a very considerable effect on the heat loss through it, and measurements of wind speed are therefore an important meteorological element to him. Many types of anemometers have been designed for such measurements.

Cup anemometers of various types are used at the great majority of stations, generally in conjunction with a chronograph which records the total number of miles of wind that pass the anemometer. The mean wind speed is of more importance to the heating engineer than the extreme speeds found in gusts. Recording wind vanes are usually employed to register the direction on the same chart, for a knowledge of the relative frequency of winds from various directions is of interest to many sections of the public.

A word regarding the exposure of wind instruments will not be amiss. Many city meteorological stations are located at airports, others (less numerous than formerly) on tall buildings. The wind in these situations, however, will be notably different from that which actually blows on the walls of houses in the residential sections of a large city. The very high frictional drag exercised on the air by a rough surface such as a city profoundly modifies the speed of the wind. It might be profitable to set up meteorological stations in

*small* parks in our cities, at least for periods long enough to evaluate the magnitude of this effect.

### Sunshine Recorders and Radiation Measurements

All the natural heat which helps the heating engineer and gives employment to his air conditioning colleagues comes from the sun, and its total amount is fairly constant when averaged over the year. Certain modern developments such as *solar* houses have greatly increased the engineer's interest in the amount of heat received directly from the sun, or from the sun and sky. The current requirement for reliable measurements of this sort is far in advance of their present number.

Measurements of the *duration of bright sunshine* have been made for about three quarters of a century, using the glass sphere which burns a track in a blue card whenever the sun is shining. This instrument is a familiar object almost everywhere in the world except in the United States of America, where a completely different sunshine recorder is in use at all principal meteorological stations. The U. S. instrument really indicates whether or not the radiation from the entire sky, including the sun, is above or below a somewhat indefinite limit.

The readings of the two types of instrument are not directly comparable, and neither is very useful to the heating engineer. Much more to his liking is an instrument (called the *pyrheliometer* in the United States, by other names elsewhere) which measures the total heat radiation received from the sun and sky on a horizontal surface. In its usual form this consists of a number of thermojunctions in series, attached alternately to a black surface and to a white one, the whole assembly being enclosed in a bulb containing dry air. The voltage developed by this instrument is recorded by an industrial recording potentiometer, permitting instantaneous readings and also hourly, daily, and monthly totals. Records of this kind have been taken for some years at more than a score of stations in the United States.

It is possible that similar records of the radiation on a *vertical* surface facing south would be of interest to heating and air conditioning engineers. To make such measurements comparable there would have to be a prior agreement on the nature of the surface below the instrument and immediately in front of it. With this limitation, the *sol-air thermometer* described by Mackey and Wright[1] should be useful. This instrument consists of an 8 in. cube of cork, covered with five spaced layers of aluminum foil, a small square on each vertical side and on the top being covered with thin cork and painted black. An embedded thermocouple measures the *sol-air temperature* of the cork, *i.e.* the temperature at which it is in equilibrium with the existing radiation, air temperature, and wind. This quantity should obviously be useful and further experience with the instrument will be of great interest.

Nocturnal radiation measurements are so much more difficult than those of the incoming radiation that they are not made anywhere as a routine, as far as the writer is aware. They have been made by skilled workers for various scientific purposes, mainly in connection with agriculture. The sol-air thermometer described may also be valuable for use at night.

### Conclusion

In this short review detailed descriptions of meteorological instruments have not been given, since this information is easily available elsewhere. It seemed better to concentrate on those general considerations of exposure which are necessary to the proper use of the instruments, and are not too familiar even to many meteorologists. It should be emphasized that the ultimate accuracy of most meteorological measurements is conditioned, not by the instruments themselves, but by the properties of the atmosphere.

[1]"The Sol-Air Thermometer—a New Instrument, by C. O. Mackey and L. T. Wright, Jr. (ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning*, May 1946, pp. 107-111.)

## FOURTH OREGON STATE AIR CONDITIONING CONFERENCE

The first Oregon State Air Conditioning Conference to be held since 1940 took place at Corvallis, Ore., March 27, 28, and 29, with an estimated crowd of more than 300 attending. Main features of the meetings were the presentation of technical papers, and the exhibition of air conditioning equipment.

The purpose of the meeting, held under the joint auspices of the Oregon Chapter of the ASHVE and the Oregon State College department of engineering, was to bring together specialists in the fields of heating, ventilating, and air conditioning, to discuss latest developments in air conditioning equipment for the benefit of those interested and to promote closer cooperation and understanding with those building trades whose products are so vitally affected by air conditioning.

Prominent among the outstanding papers presented at the conference by members of the Society were: Duct Design and Air Distribution by Walter Hanthorn, Kleer-air Furnace Co.; Ventilation and Exhaust by C. R. Gelert, president, Gelert Co.; Contractors' Slant on Panel Heating by W. J. Widmer, Widmer Plumbing & Heating Co.; Cooling and Dehumidifying by J. A. Freeman, Western Engineering Co.; Split Systems by T. H. McClung, Brod & McClung; Controls by J. S. Locke, Minneapolis-Honeywell Regulator Co.; the Heat Pump by W. E. Johnson, General Electric Co.

On the second day of the conference W. Bruce Morrison, president of the Oregon chapter during 1946, gave a resume of the past year after which Tom E. Taylor, past president, Oregon chapter, installed the new officers for 1947.

Officiating as chairmen of the various sessions were: J. D. Kroeker and T. E. Taylor, consulting engineers; W. J. Kollas, the Lynch Co.; F. E. Urban, Urban Plumbing & Heating Co.; W. Bruce Morrison, consulting engineer; W. H. Stover, General Electric Co.; W. C. Vradenburg, North Coast Engineers; B. W. Farnes, Perfex Corp.; and C. W. Brissenden, newly elected president of the Oregon chapter.

# Coronado Awaits ASHVE

## FOR JUNE MEETING

PLANS for the Semi-Annual Meeting of the Society at Coronado, Calif., June 2-4, 1947, are shaping up nicely under the direction of R. A. Lowe, general chairman, Committee on Arrangements.

On behalf of the Officers of the Southern California Chapter, Pres. Art Theobald extends a cordial welcome to all members and suggests that business and pleasure be combined in attending the meeting at the Hotel del Coronado, where an interesting program of technical sessions will be held. There will also be an opportunity for inspection trips, sightseeing and other entertainment events which will be available for members, their wives and children.

A special bulletin has been sent to members by mail with application forms for transportation, hotel reservations, and other special events. These should be mailed promptly upon receipt to the proper person.

Members making the trip by rail should consult local railroad representatives for information about special cars, and should also select their return route, as there are many options that will provide access to scenic and historic areas throughout the country. Visits can be made to the national parks such as Grand Canyon, Zion, Yosemite, Glacier, Yellowstone, Estes, etc.

Quick airline service is available to San Diego and limousine service is available from the airport to Coronado.

Motorists can find many interesting points to visit en route to the meeting. Automobile clubs will furnish up to date information on the best routes. After arrival in San Diego, the trip to Coronado requires a ten minute ferry ride across the bay. Daytime service is provided every ten minutes and after midnight every half hour.

When making hotel reservations it is desirable to indicate the estimated time of arrival and period of your stay.

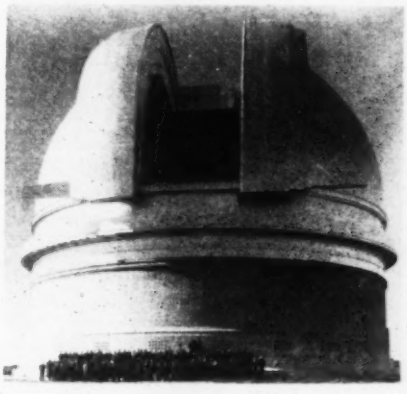
For those who contemplate a visit to Mexico, the United States Immigration Office advises that if a member's birthplace is outside of the United States, proof of citizenship is necessary. Canadian citizens visiting Mexico should have, in addition to their Canadian passport, a visitor's visa, Section 3-2, obtainable from their American Consulate.

Members who were born in Canada or any other foreign country, who may now hold American citizenship papers, should have such citizenship papers certified by Citizenship Form No. 1993, or a voter's certificate to insure reentry into the United States.

United States Customs Office advises that all purchases in Mexico require payment of duty. Duty will be collected providing the purchaser's stay in Mexico is less than 24 hours. Prohibited articles are sugar, liquor, narcotics.

All foreign-made merchandise must be declared before crossing the border into Mexico, otherwise duty will be charged on the return trip.

**Mt. Palomar is the site of the world's largest observatory telescope**



For the technical sessions the Meetings Committee has provided an interesting group of papers, and it should be noted that three morning sessions will be held June 2-4. The first session will be devoted to the presentation of research papers; the second session to subjects on use of climatological data, air sterilization, and design temperatures; the third session will provide papers on comfort conditions and radiant heating and cooling studies.

It will be of assistance to the Committee on Arrangements if members will use the special forms provided and register for the various inspection and sightseeing trips, golf tournament, and ladies' events. For the first time since 1942, the Research Cup and the Eichberg Memorial Cup will be in competition for the golfers who will play at the San Diego Country Club.

The Semi-Annual Banquet will be held on Tuesday evening, June 3, and the featured speaker will be Dr. Stafford Warren, dean of the Medical School, University of California, in Los Angeles, whose subject will deal with the physiological effects of atomic energy through the channels of water supply, vegetation, contamination of materials and air-borne effects. Dr. Warren headed the medical staff under General Leslie Groves during the war. Following the banquet in the main dining room there will be dancing in the Circus Room.

At the conclusion of the meeting it is anticipated that many members will go to Los Angeles to attend the Spring meeting of the *American Society of Refrigerating Engineers*, which will be held June 9-11 at the Alexandria Hotel.

Come along to Coronado and combine business and pleasure where the sun shines all of the time.

JOURNAL SECTION

# Program, Semi-Annual Meeting 1947

## American Society of Heating and Ventilating Engineers

### Sunday, June 1

10:00 a.m. Registration—Lobby, Hotel del Coronado
2:00 p.m. Council Meeting—Reading Room
7:00 p.m. Nominating Committee

### Monday, June 2

9:00 a.m. Registration—Lobby
10:00 a.m. RESEARCH SESSION—Ball Room
- Challenge of the ASHVE Research Program, by Cyril Tasker
- The Thermo-Physical Properties of Building Materials, by H. B. Nottage
- Heat Transfer by Forced Convection Along a Smooth Flat Surface, by G. V. Parmelee and R. G. Huebscher
- Influence of Gaseous Radiation in Panel Heating, by F. W. Hutchinson

10:30 a.m. Children's Sports—Playgrounds
12:30 p.m. Luncheon
12:30 p.m. Deep-Sea Fishing Trip
2:00 p.m. Roads to Romance—Old Town—Ramona's Wedding Place—Balboa Park. (Buses leave West Entrance of Hotel)
- Golf Tournament—Research Cup and Eichberg Memorial Cup, San Diego Golf Club
- Beach Party for Children

6:30 p.m. Dinner—Main Dining Room
8:30 p.m. Chapter Delegates' Meeting—Gold Room
Movies

### Tuesday, June 3

9:00 a.m. Registration—Lobby
9:30 a.m. TECHNICAL SESSION—Ball Room
- Conditions for Comfort, by C. S. Leopold
- A Method of Improving the Effective Temperature Index, by C. P. Yaglou
- Physiological Adjustments of Human Beings to Sudden Change in Environment, by Nathaniel Glickman, Tohru Inouye, Stanley E. Telser, M. D., Robert W. Keeton, M. D., Ford K. Hick, M. D., and Maurice K. Fahnestock
- Comfort Reactions of 275 Workers During Occupancy of Air Conditioned Offices, by F. B. Rowley, R. C. Jordan and W. E. Snyder.

10:30 a.m. Ladies' Card Party
12:30 p.m. Luncheon
2:00 p.m. Harbor Cruise and Inspection of Navy Yards. Leave Hotel Boat House
5:30 p.m. Social Hour—Swimming Pool Terrace
7:00 p.m. Banquet—Main Dining Room
- *Toastmaster:* Leo Hungerford
- *Speaker:* Dr. Stafford Warren, Dean, Medical School, University of California, Los Angeles
- *Subject:* The Significance of the Bikini Tests to Engineering

Dancing in Circus Room

### Wednesday, June 4

9:30 a.m. TECHNICAL SESSION—Ball Room
- Experimental Studies on Panel Heating Tube Spacing, by B. F. Raber and F. W. Hutchinson
- Effect of Floor Slabs on Building Structure Temperatures and Heat Flow, by C. F. Kayan
- Triethylene Glycol Vapor Distribution for Air Sterilization, by Edward Bigg, B. H. Jennings and F. C. W. Olson
- Determination of Outside Design Temperatures, by W. L. Holladay

12:30 p.m. Luncheon
2:00 p.m. Trip to Old Mexico, the Land of Manana. Buses leave West Entrance of Hotel.

### Method of Choosing Location of, Financing and Conducting Meetings of the Society

*Resolved:* That inasmuch as the Annual and Semi-Annual Meetings of the Society come under the jurisdiction of the Council, the following rules governing the handling of such meetings be adopted by the Council and published in the JOURNAL of the Society at least twice during every year, preferably just prior to each meeting.

1—The Council will select the city in which the Annual or Semi-Annual Meeting is to be held, giving due consideration to the invitations received from Chapters or members as well as to the advisability of so distributing those meetings as to make them of the greatest advantage to the general membership, and to reduce as far as possible the expense of members attending.
2—That an appropriation be made to cover the entertainment or local expenses, incurred in connection with the meeting, not exceeding $500.00, the regular meeting expense to be taken care of by the General Fund of the Society in the regular way.
3—That no registration fee or compulsory, obligations of any nature be imposed on members or guests.
4—That the purchase of tickets for banquets or for any other form of entertainment that may be provided be entirely voluntary.
5—That the grouping of features and the sale of tickets for group features be discouraged.
6—That the raising of funds from manufacturers of heating apparatus be discouraged.
7—That the display of samples, or of literature, advertising the product of any manufacturer in any way, shape or form, be not permitted at the booths, registration desk, or in or about the meetings.
8—That the distribution of trade papers be entirely at the discretion of the committee in charge.
9—That the local Chapter or local members be empowered to form a General Committee with such sub-committees as may be required to handle the details of transportation, hotel accommodations, entertainment, finance, etc., and that this General Committee be requested to confer frequently with the Council, through the Secretary of the Society, and to make frequent reports on progress in connection with the various matters being handled by them.
10—That the arrangements of elaborate and costly entertainment features be discouraged.

*Adopted at Council Meeting, January 29, 1926.*

## SUGGESTED ITINERARY

For the convenience of members in various cities who may wish to travel with a group of their follow members a suggested itinerary has been prepared and arrangements have been made for a railroad representative in each city to make transportation arrangements. Members desiring to use this service can call the nearest representative listed on the reservation form and plan their trip to the Semi-Annual Meeting at Coronado.

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|---|---|---|---|---|
| Lv. | Montreal | May 28 | 11:00 p.m. | C. N. |
| " | Boston | May 29 | 8:01 a.m. | New Haven |
| " | Toronto | " | 8:00 a.m. | C. N. |
| " | New Haven | " | 11:15 a.m. | New Haven |
| " | New York | " | 2:35 p.m. | B. & O. |
| " | Philadelphia | " | 5:08 p.m. | B. & O. |
| " | Washington, D. C. | " | 8:00 p.m. | B. & O. |
| " | Syracuse, N. Y. | " | 5:00 p.m. | N. Y. C. |
| " | Buffalo | " | 8:20 p.m. | N. Y. C. |
| " | Winnipeg | " | 7:00 p.m. | G. N. |
| " | Detroit | " | 11:50 p.m. | B. & O. |
| " | Atlanta | " | 7:25 p.m. | West Pt. Route |
| " | Pittsburgh | " | 10:10 p.m. | B. & O. |
| " | Cleveland | May 30 | 12:15 a.m. | N. Y. C. |
| " | Columbus | " | 8:35 a.m. | Penn. |
| " | Cincinnati | " | 9:05 a.m. | B. & O. |
| " | Indianapolis | " | 11:05 a.m. | Penn. |
| " | Chicago | " | 11:00 a.m. | R. I. |
| " | Minneapolis | " | 11:30 a.m. | R. I. |
| " | St. Louis | " | 4:00 p.m. | Mo. Pac. |
| " | Des Moines | " | 4:55 p.m. | R. I. |
| " | Omaha | " | 4:45 p.m. | M. P. |
| " | Kansas City | " | 10:50 p.m. | R. I.—S. P. |
| " | Memphis | " | 8:10 a.m. | M. P. |
| " | Oklahoma City | " | 8:15 a.m. | Santa Fe |
| " | New Orleans | " | 11:30 a.m. | S. P. |
| " | Seattle | " | 3:50 p.m. | S. P. |
| " | Denver | " | 7:50 p.m. | Santa Fe |
| " | Houston | " | 8:10 p.m. | S. P. |
| " | Dallas | " | 9:15 p.m. | T. & P. |
| " | Ft. Worth | " | 10:00 p.m. | T. & P. |
| " | Portland | " | 10:00 p.m. | S. P. |
| " | San Antonio | May 31 | 1:00 a.m. | S. P. |
| " | Salt Lake City | " | 9:45 a.m. | U. P. |
| " | El Paso | " | 5:20 p.m. | S. P. |
| " | San Francisco | " | 9:00 p.m. | S. P. |
| " | Los Angeles | June 1 | 11:30 a.m. | Santa Fe |
| Arr. | San Diego | " | 3:15 p.m. | |

R.R. time tables subject to change April 27. Should be checked when filing reservations.

# JOURNAL SECTION

## COMMITTEE ON ARRANGEMENTS

R. A. LOWE, *General Chairman*

J. L. BLAKE — LEO HUNGERFORD
R. S. FARR — MARON KENNEDY

*Committee Chairmen*

*Finance*.....W. N. PAULEY
*Hotels and Reservations*.....A. G. OREAR
*Industry*.....L. B. DAVENPORT
*Inspection Trips*.....MARON KENNEDY
*Ladies*.....A. B. BANOWSKY
*Outside Entertainment*.....J. S. EARHART
*Reception and Publicity*.....E. A. GUENTHER
*Special Events*.....VERN MILLER
*Sports Events*.....BOB JOHNSON
*Transportation* (Local).....C. W. POLLOCK

## SOUTHERN CALIFORNIA CHAPTER OFFICERS

*President*.....Art Theobald
*Vice President*.....R. A. Lowe
*Treasurer*.....J. L. Blake
*Secretary*.....R. S. Farr

*Board of Governors*

Maron Kennedy — J. S. Earhart
L. B. Davenport — J. M. Moriarty

**Business and pleasure may be combined in attending the Semi-Annual Meeting at the Hotel del Coronado**

JOURNAL SECTION

# Summary of Local Chapter Meetings*

ATLANTA—March 7, 1947. Subject: Human Engineering. Speaker: Lewis Gordon, vice president, Citizens and Southern National Bank. Mr. Gordon gave an interesting talk which the members enjoyed. Other Features: The business meeting was called to order by Pres. H. King McCain. G. R. Bilderback reported on the chapter's finances and W. W. Carder reported on future programs. L. L. Barnes announced that 25 per cent of the chapter members had pledged contribution to the fund for reducing the mortgage on the Society's Research Laboratory building. Attendance 53. Attendance ratio 0.52.

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CENTRAL OHIO—March 17, 1947. Subject: Heat Transfer to and from Direct Fired Heating Surfaces. Speaker: Prof. Lorin G. Miller, head, mechanical engineering dept., Michigan State College, East Lansing, Mich. Professor Miller opened his remarks with a review of the transfer of heat through materials and the laws that govern the rate of heat transfer. He showed slides to illustrate how much greater the heat transfer was when surfaces were operated as near as possible to the temperatures at which they begin to scale. By taking advantage of high temperatures to get increased rates of heat transfer, Professor Miller showed how it was possible to design a furnace of minimum size and still have one of durability. Professor Miller presented his subject in an interesting manner and held the attention of his audience to the end. Considerable discussion followed. Attendance 50. Attendance ratio 0.40.

CENTRAL NEW YORK—March 19, 1947. Subject: Economic Aspects of the Use of Copper and Copper Alloys. Speaker: R. A. Wilkins, vice president in charge of research, Revere Copper and Brass, Inc. Mr. Wilkins gave an informative talk, which was followed by two films, one on fabrication of copper and copper alloy products and the other on fabrication of magnesium. Seeing the actual production and reviewing it in motion picture resulted in great enthusiasm. Preceding the talk there was an inspection trip through the mills of the Rome Division of Revere Copper and Brass. Efficient guides conducted the members and guests through the office, where interesting operations, including extrusion of pipe, were observed. Before the meeting was adjourned the members observed two minutes of silence in tribute to Theodore S. Steinhorst, a chapter member from Utica, N. Y., who had died since the last chapter meeting. Attendance 32.

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CINCINNATI—March 4, 1947. Subject: Invention and Development of Heat Pump Equipment. Speaker: Norman R. Kevers, president, Terra-Temp, Inc., Indianapolis, Ind. Mr. Kevers' description of the research involved and of the operation of the equipment was amusing as well as instructive. A discussion followed. Other Features: R. W. Sigmund reported as chairman of the membership committee and announced that there was a net gain of 11 members in recent months with a total of 63 on the Chapter's roll. The nominating committee reported that the nominees had been selected and would be submitted at the next meeting. Attendance 39. Attendance ratio 0.24.

CINCINNATI—February 4, 1947. Subject: Radiant Heating. Speakers: Carl C. Weber, A. M. Byers Co., Pittsburgh, Pa.; R. J. Vanderweil, Chase Copper and Brass Co., Waterbury, Conn. Each of the speakers gave a brief talk on practical phases of radiant heating installations, followed by a general discussion, with the two speakers on the receiving end of a barrage of questions from the group. This was both educational and entertaining. Other Features: Pres. A. W. Edwards gave a brief talk on the activities at the Society's Annual Meeting in Cleveland. R. W. Sigmund reported for the membership committee and W. J. Killian reported for the entertainment committee. Attendance 75. Attendance ratio 0.47.

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CONNECTICUT—March 13, 1947. Subject: Heat Flow in Building Structures by Electrical Analogy. Speaker: Carl F. Kayan, assistant professor, mechanical engineering, School of Engineering, Columbia University, New York. A discussion followed in which suggestions were offered and many questions asked. Other Features: Pres. Paul D. Bemis, reporting for the legislative committee chairman, W. K. Simpson, stated that this committee was active in attending the hearings on the Connecticut State Registration Board and Senate Bill 545 in regard to requirements of boiler operation on high pressure systems. A. J. Lawless reported for the meetings committee. Allan Hubbard, in reporting for the special chapter committee on consulting engineers' fees, announced that a formal meeting with architects in the State would be held in the near future. President Bemis announced that the Connecticut Chapter to date had contributed $550.00 toward the fund to retire the mortgage on the Society's Research

*Note: The attendance ratios shown represent the membership attendance divided by the chapter membership. These ratios will be useful as a partial indication of interest shown by local chapter members in various types of subjects programmed by the various chapters and may be useful in deciding on subjects for chapter meetings.

Laboratory building in Cleveland. Attendance 70. Attendance ratio 0.35.

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DELTA—March 11, 1947. Subject: Locker Freezing Plants. Speaker: Clayton V. Bankston, district representative, Industries Sales Corp., New Orleans, La. At the conclusion of his talk, Mr. Bankston invited questions from the floor and an interesting discussion followed. Other Features: A testimonial to the memory of John Devlin was read by Cary B. Gamble. Mr. Devlin was a charter member of the Delta Chapter and was one of its most beloved members. Following the reading of the testimonial, the following resolution was adopted:

Be it Resolved, That in behalf of the entire membership of Delta Chapter of AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS this testimonial be adopted in tribute to the memory of our late fellow member John Devlin. Be it Resolved, That this testimonial be recorded in the minutes of the Society, and a copy be sent to his wife, son, and brothers, with an expression of sympathy and heartfelt condolence.

The resolution was submitted by L. K. Nelson, L. V. Cressy and C. B. Gamble. Attendance 24.

The February meeting was a stag buffet supper meeting and no business was transacted.

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GOLDEN GATE—March 6, 1947. Subject: Design, Application and Results Obtained with Multi-Vent Panels as Applied to Heating and Air Conditioning Installations. Speaker: Mel J. Stevenson, consulting mechanical engineer, The Pyle National Co., Industrial Multivent Div., Chicago, Ill. His talk was accompanied by charts which he used in the progress of his talk. A lengthy discussion followed. Other Features: Pres. James Gayner announced the appointment of the nominating committee as follows: R. B. Holland, chairman; C. L. Peterson and J. I. Sprott. Fred W. Kolb, chairman of the program committee, announced the arrangements for the future meetings. Paul R. Babcock, chairman of the technical committee, reported that the revision of the code had been presented to the Board of Supervisors and that, as yet, no decision had been reached on the recommendations. President Gayner announced that the technical committee also assists in outlining and in the selection of programs for the Chapter meetings and passes on the talks before they are accepted. Attendance 81. Attendance ratio 0.67.

GOLDEN GATE—February 6, 1947. Subject: Theory and Practical Design of Air Filters. Speaker: William Pauley, chief engineer, Farr Co., Los Angeles, Calif. He outlined the use of filters, the type of filtering applications, and the various types of filters. He also discussed the fundamental considerations in the testing of filter performance. Many questions were asked in the discussion that followed and these were answered to the satisfaction of the members. Other Features: C. E. Bentley, delegate to the Annual Meeting in Cleveland, gave his report and also announced that Neil H. Peterson, Golden Gate Chapter member, has been elected chairman of the Society's Nominating Committee. He also announced that the Semi-Annual Meeting of the Society was to be held at Coronado, Calif., June 1-4, 1947. John Everetts, Jr., alternate to the Annual Meeting, also gave a report on the activities at the meeting. F. L. Nemic, member of the technical committee, reported that they were getting together suggested recommendations covering the ventilation of buildings, and also they were recommending that the section on vents and stacks be eliminated, and that the code of the Board of Underwriters covering same be substituted, but, as yet, no decision had been reached. However, he mentioned that progress was being made. Mr. Bentley suggested that any members having ideas or suggestions relative to the revision of the code confer with the technical committee. Attendance 56. Attendance ratio 0.66.

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ILLINOIS—March 10, 1947. Subject: Recent Trends in Safety Control in Ventilating Systems. Speaker: J. T. W. Babcock, assistant chief engineer, *Factory Insurance Association*, Chicago, Ill. Mr. Babcock reviewed many of the details of ventilating design, installation, and maintenance that should be given attention to reduce the fire hazard. He called attention to two proposed changes in *NFPA* regulations, one involving a ruling against the use of corridors as return air passages and the other permitting the design of the ventilating or air conditioning systems for the removal of smoke. These changes, he stated, are merely suggestions and are not, as yet, in effect. Other Features: C. W. Johnson, chairman of the board of teilers, announced that Messrs. A. O. May, W. A. Kuechenberg, J. S. Kearney, C. E. Crone, J. J. Philippi, P. J. Marschall and A. L. Crump were elected to serve on the chapter's nominating committee. J. F. Cummiskey, chairman of the meetings committee, announced that Dr. Baldwin M. Woods, president of the Society, Berkeley, Calif., was to be the guest speaker at the April meeting. Attendance 82. Attendance ratio 0.20.

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MANITOBA—March 20, 1947. The March meeting was held at the Fort Garry Hotel with D. S. Swain presiding. The minutes of the January meeting were adopted on motion of D. G. Saunders, seconded by A. W. Moss. Einar Anderson, delegate to the Society's Annual Meeting in Cleveland, reported on the activities. D. F. Michie was appointed chairman of a committee to write a history of the Manitoba Chapter. Messrs. Michie, Moss and R. L. Kent were appointed to serve on the nominating committee. Three interesting films of current topics were shown at the conclusion of the business session.

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MASSACHUSETTS—March 18, 1947. Subject: Installation Troubles with Fan Systems. Speaker: John Cassie, mechanical engineer, Westinghouse Electric Corp., Sturtevant Div., Hyde Park, Mass. Following

the talk the secretary gave a report on the fund being raised by the Massachusetts Chapter toward paying off the mortgage of the Society's Research Laboratory building in Cleveland. Attendance 65. Attendance ratio 0.80.

MASSACHUSETTS — February 18, 1947. Subject: District Heating Moves Forward. Speaker: G. K. Saurwein, mechanical engineer, New England Gas and Electric Co., Cambridge, Mass. Other Features: Pres. C. W. Larson gave a report on the fund being raised by the Chapter for paying off the mortgage on the Research Laboratory building and it was voted to solicit all members who had not yet contributed. Attendance 42. Attendance ratio 0.54.

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MICHIGAN—March 17, 1947. Subject: Room Air Distribution—Science or Guess Work. Speaker: George L. Tuve, 1st vice president of the ASHVE and professor of mechanical engineering, Case School of Applied Science, Cleveland, Ohio. The material for his talk was taken principally from results obtained in the research project which Case is carrying on in cooperation with the Society at its Research Laboratory on the action of free air streams after leaving outlets and openings in duct systems. Professor Tuve's talk was well received and a rising vote of thanks was extended to him. Other Features: M. B. Shea, long time member and a past president of the Michigan Chapter, who has been retired by the American Radiator Co. and to whom the meeting was dedicated, was presented with a pocket case and a framed scroll in appreciation for the many years of loyal service he has given the Chapter. Mr. Shea, who is leaving Detroit to live with a daughter in Pennsylvania, was given a hearty farewell with the best wishes of the Chapter to go with him. Pres. F. R. Bishop made the presentation to Mr. Shea. President Bishop read an announcement for contributions to the fund for reducing the mortgage on the Society's Research Laboratory building in Cleveland. Attendance 102. Attendance ratio 0.65.

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MINNESOTA—March 4, 1947. Subject: Air Distribution. Speaker: Ernest Szekely, president, Bayley Blower Co., Milwaukee, Wis., and member of the Council of the Society. Mr. Szekely gave an interesting talk and illustrated his remarks with slides. Other Features: Pres. J. E. Haines reported briefly on the activities at the Annual Meeting in Cleveland. C. T. Lawrence reported on the contributions to the fund to reduce the mortgage on the Society's Research Laboratory building and announced that $591.00 had been collected to date. He also announced that the fund would be kept open until June. The following were elected to serve on the nominating committee: A. B. Algren, R. E. Gorgen, William McNamara, William Sturm and W. F. Uhl. Lyman Gross introduced the new president of the *Minnesota Federation of Engineering Societies*, Paul R. Speer, who spoke briefly. He in turn introduced the retiring Federation president, S. L. Stolte, who gave a resume of the Federation's activities during the past year. Attendance 71. Attendance ratio 0.39.

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MONTREAL—March 17, 1947. Subject: Metering and Combustion Control Equipment. Speaker: Lew Stewart. He gave a very interesting and instructive talk, which he illustrated with slides. As reported by the secretary, few questions were asked due to the fact that the talk was so ably presented and carefully prepared. Secretary Salter stated that Mr. Stewart is, no doubt, one of the best qualified men in that section of the country on the subject of controlled costs and equipment efficiency. Other Features: The meeting was preceded by the annual curling bonspiel which was held at the Caledonia Curling Club. A discussion on the research mortgage fund was headed by F. G. Phipps and supported by retiring President Madden. V. J. Jenkinson, president of the Ontario Chapter, was introduced and he gave his ideas on the proposed joint meeting between the Montreal, Ontario and Western New York Chapters, which is now scheduled for the Fall. Attendance 66.

MONTREAL—February 17, 1947. Subject: Radiant Heating and its Application. Speaker: G. Lorne Wiggs, consulting engineer, Montreal, Que., Canada. The popularity of this subject was proved by the second largest attendance in the history of the Montreal Chapter. Mr. Wiggs showed a series of slides and performance curves. Other Features: Retiring president, A. B. Madden officially presented the incoming president, T. H. Worthington, who expressed his thanks to Mr. Madden and complimented the past president on his fine record of accomplishment during his term of office. President Worthington announced the various members of the committees who will serve during the coming year. Mr. Madden presented his report on the happenings at the Annual Meeting. He stressed the fact that it is most desirable that the Chapter delegate be thoroughly familiar with the ASHVE procedure and Chapter problems before attending Society meetings. Messrs. Garneau and Plamondon were given a vote of thanks for showing the colored travelogue movie through the courtesy of the Provincial Government. Attendance 70.

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NEBRASKA—March 26, 1947. Subject: Prospective Change-Over and Ramifications of 600 Btu Manufactured Gas to 1000 Btu Natural Gas for the City of Omaha. Speaker: W. B. Manchester, Metropolitan Utilities District and president of Omaha Engineers' Club. Mr. Manchester's talk was very timely and caused a series of questions mostly regarding gas rates per 1000 Btu. Attendance 24.

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NEW YORK—March 17, 1947. Subject: Air Conditioning in the Development of New Industries. Speaker: Capt. A. E. Stacey, Jr., second vice president of the Na-

tional Society, and process consultant of the Carrier Corp., Syracuse, N. Y. Captain Stacey pointed out that in the field of air conditioning engineers must know both the old problems and the new, since many of the seemingly new developments actually are refined applications of known fundamentals. That this presentation was both timely and informative was shown by the lively discussion period that followed. Other features: Past president C. S. Pabst presented to Harold J. Ryan in the name of the Chapter the past president's gift, which was a barometer bearing the ASHVE emblem. Mr. Pabst spoke of the faithful and enthusiastic service Mr. Ryan had given the chapter in the past year and also expressed the gratitude of the chapter to Mr. Ryan. In accepting the gift Mr. Ryan spoke of his great enjoyment in serving the chapter in the pleasant association of the other officers and various committee members who had contributed to the success of the chapter. Attendance 81.

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NORTH TEXAS—March 17, 1947. Subject: Programs Under Way at the Texas State Research Foundation. Speaker: Dr. D. R. Carmody, Texas State Research Foundation. Dr. Carmody delivered a well planned outline of the various programs which the Foundation now has under study, specifically, (1) Agricultural; (2) Industrial; (3) Medical; and (4) Miscellaneous, including special research for the petroleum and related industries. Dr. Carmody mentioned that the Foundation specializes in conducting studies for the benefit of small industries. Other Features: Pres. C. Rollins Gardner named B. S. Foss, Jr., as delegate to the Semi-Annual Meeting in Coronado and to the next Annual Meeting in New York in 1948. The Texas A & M Student Placement Service was described and the members urged to take advantage of this service in securing worthwhile employees for their organizations. Attendance 44. Attendance ratio 0.43.

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NORTHERN OHIO — March 10, 1947. Subject: Radiant Disinfection of Air. Speaker: Prof. W. F. Wells, director, Laboratory for Study of Air-Borne Infection, University of Pennsylvania Medical School, Philadelphia, Pa. Professor Wells urged all ventilating engineers to investigate the latest developments in germicidal lamps, and to acquaint themselves with statistics now available. He showed slides which indicated areas reached by various sizes and types of lamps. Professor Wells quoted from records indicating a reduction in air-borne bacteria wherever this equipment is installed. An interesting discussion followed. Other Features: Pres. J. E. Wilhelm introduced H. G. Dyktor, Industrial Hygiene Division, who stressed the importance of installing proper equipment for contaminant removal. He also mentioned that a code is now in effect governing this work. W. R. Moore reported that the Mortgage Retirement Fund had reached $1520.00 and urged the members to try for a new goal of $2,000.00. Don Taze gave his final report on the Annual Meeting and the Chapter expressed its appreciation for the excellent job he did as general chairman. Attendance 84. Attendance ratio 0.53.

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OKLAHOMA — March 10, 1947. Subject: Acoustics as Applied to Air Conditioning. Speaker: H. O. McElyea, regional manager, construction department, Johns-Manville Sales Corp., St. Louis, Mo. A general discussion followed Mr. McElyea's interesting talk. Attendance 17.

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OREGON—March 28, 1947. Annual Meeting held in conjunction with the Fourth Oregon State Air Conditioning Conference. Subject: The Shortest Distance Between Two Points is Not a Straight Line. Speaker: L. B. Prillaman of the Lincoln High School Staff. Mr. Prillaman's talk was an instructive lesson on geography, international intrigue and communism. A history of previous conferences was given by Earl C. Willey. Visitors from California and Washington were introduced and Thomas E. Taylor unveiled the welcome sign. Dr. A. L. Strand, president of Oregon State College extended greetings. The Southern California Chapter extended an invitation to the Summer Meeting, June 1-4. Exhibit chairman Thomas E. Taylor instructed exhibitors on shipping arrangements for their equipment. President Morrison reviewed the accomplishments of the various committees during 1946-7 and cited proof of their effective work and cooperation in the fact that the approved applications for new members will show a gain of 70 per cent for the fiscal year. The committees were thanked and discharged. Past president Thomas E. Taylor introduced new officers and presented the chapter gavel to President Brissenden. The other new officers are: Vice President E. R. Lokey; Treasurer, R. C. Chewning; Secretary, L. G. Williams; Board of Governors, W. J. Kollas, W. R. Norte, and W. B. Morrison; Alternate Delegate, C. W. Brissenden. President Brissenden then presented an engraved gavel to the retiring president, W. B. Morrison. Attendance 187.

OREGON—March 6, 1947. Subject: Atom Tests. Speaker: Jack Dement, scientist and lecturer, Portland. Mr. Dement, the youngest of the 21 scientific observers to be selected by the War and Navy Departments to observe the Bikini atom test, explained many little known facts and told of the tremendous heat loss in the separation of the required metals. Following Mr. Dement's interesting talk, Dr. H. C. Dake, editor, Mineralogist Magazine, demonstrated the Geiger counter and various rare metals. Other Features: Article B, Section 8 of the By-Laws was referred to the legislative committee for action. The nominees for the 1947-48 officers were unanimously elected as follows: *President*—C. W. Brissenden; *Vice President*—E. R. Lokey; *Treasurer*—R. C. Chewning; *Secretary*—L. G. Williams. *Board of Governors:* W. J. Kollas, W. R. Norte and W. Bruce Morrison. Attendance 69. Attendance ratio 1.11.

# JOURNAL SECTION

PACIFIC NORTHWEST—March 11, 1947. Subject: Production, Distribution, Engineering, Consumption and Control of Central District Steam. Speakers: Alex Lux, Puget Sound Power and Light Co., Central District Steam Div., spoke on production and distribution; C. W. May, consulting engineer, Seattle, spoke on engineering; W. B. Pride, heating and ventilating engineer, Boeing Aircraft Co., Seattle, spoke on consumption; and S. D. Peterson, branch manager, Johnson Service Co., Seattle, discussed the control problems encountered. Other Features: E. H. Langdon, delegate to the Annual Meeting in Cleveland, told of the activities at the meeting and placed particular emphasis on the matter of donations by the various chapters to the Research Retirement Fund. He pointed out that they are planning on doing more and more research work at the Laboratory in Cleveland which will be for the benefit of all connected with the industry and which will justify all the time, effort and money spent for research. Mr. Peterson was appointed to represent the chapter at the Oregon State Air Conditioning Conference at Corvallis, Ore. Attendance 37.

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PHILADELPHIA—March 13, 1947. Subject: Fan Selection. Speaker: R. S. R. Reade, Jr., American Blower Co., Detroit, Mich. A discussion followed. Other Features: The treasurer's report was read and approved. A resolution was read by A. M. Robertson in memory of Robert C. Morgan, life member, who passed away January 30th. Attendance 83. Attendance ratio 0.65.

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PITTSBURGH—March 10, 1947. Subject: Ways and Means of Decommissioning Ships. Speaker: Prof. E. R. Queer, professor of Engineering Research, Pennsylvania State College, State College, Pa., and former Commander in the U. S. Navy. Professor Queer was actively engaged in the planned Navy program of decommissioning ships, and after hostilities ended, the program was started in order to preserve Naval ships from deterioration. He explained the complete process in detail and then showed movies which the Navy uses in training personnel in the application of the process. Other Features: Pres. A. F. Nass spoke in regard to chapter news and then announced the coming Semi-Annual Meeting scheduled for Coronado, Calif., June 1-4. John F. Collins, Jr., submitted the report of the nominating committee. D. W. Loucks gave his report as chapter delegate to the Annual Meeting. President Nass then appointed the following to write up the history of the Pittsburgh Chapter: John F. Collins, Jr., H. Lee Moore and P. A. Edwards. Attendance 42. Attendance ratio 0.48.

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ROCKY MOUNTAIN—March 5, 1947. Subject: Controls of the Future. Speaker: Gordon Volkenant, Minneapolis-Honeywell Regulator Co. The secretary reported one of the most interesting talks of the year. Other Features: J. F. Mohan read the proposed amendment to Section 3, Article R-5, of the Chapter's By-Laws and a motion was made and seconded that the following be adopted: "Twenty per cent of the regular and associate members in good standing, as evidenced by the Secretary's membership record, shall constitute a quorum for the conduct of all regular and special meetings." Attendance 49.

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SOUTH TEXAS—March 21, 1947. Subject: Trends in Duct Layouts and Design. Speaker: George E. May, principal utility engineer, New Orleans Public Service Inc. Mr. May gave an excellent, informative talk, after which the meeting was thrown open to discussion. Other features: Bert Fisher introduced the past president, and the present president of the A & M student branch at College Station who advised members and their wives to attend the April 17th meeting at College Station. Mr. Wayne Long reported on plans for a short course at A & M college which will be held June 8-11. Arthur Barnes, reporting on the meetings of the Engineers Council, advised they are making a study of the underground construction of Houston from the standpoint of locating and identifying this construction. Mr. Barnes also raised the question of the city code, and Mr. Fisher advised that no meetings have been held lately since they are waiting for copies of codes from other cities before proceeding further with their study. A motion made to continue meetings through the summer was carried with one dissenting vote. The committee on entertainment presented the Hot Air Recorder fund with $100.00 from their surplus fund. President Bert Fisher appointed a Classification Committee whose duties will be to classify all applicants for membership and make recommendations to the National Society. Attendance 99.

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SOUTHWEST TEXAS—March 11, 1947. Subject: Wrought Iron Pipe from Standpoint of Radiant Heating Installations. Speaker: A. D. Sheers, division manager, A. M. Byers Co., Houston, Tex. Following Mr. Sheers' talk a motion picture, *Eternally Yours*, was presented and enjoyed. Following the showing of the film many questions involving wrought iron pipe were asked. Three members from the South Texas Chapter and three members from the Student Branch at Texas A & M College were among the visitors. Attendance 22.

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UTAH—March 10, 1947. Subject: Responsibility of the Heating Engineer to the Community. Speaker: W. L. Butler, chief engineer, Power and Heating Division, Salt Lake City. Mr. Butler enumerated the responsibilities of the professional heating engineer. He also outlined the accomplishments of the Power and Heating Division in recent years by noting that there has been a consistent decline in heating plant fires, a reduction in heating plant failures, and an improvement in commercial heating plants throughout the city. With respect to the smoke problem in Salt Lake City,

Mr. Butler stated the solution is mechanized firing equipment. The interest in Mr. Butler's talk was evidenced by the numerous questions asked by those present. Other Features: The mortgage retirement plan to clear the mortgage on the Society's Research Laboratory building in Cleveland was discussed. The newly elected officers were announced as follows: President—E. V. Gritton; Vice-President—E. J. Watts; Secretary-Treasurer—C. E. Ferguson. Attendance 20. Attendance ratio 0.50.

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WASHINGTON, D. C.—February 15, 1947. Subject: Commercial Aircraft. Speaker: W. B. Shippen, Glenn L. Martin Co. Mr. Shippen outlined the salient features now being built into commercial aircraft for maintaining habitability, and discussed some of the ways that the unusual problems presented by high speed, high altitude commercial airplanes are being solved. A lively discussion followed and during this period Mr. Shippen further demonstrated his familiarity with special difficulties seldom encountered except by the aviation engineer. Other Features: Pres. L. Bert Nye, Jr., congratulated A. E. Beitzell on the success of the inspection trip which he had arranged at the National Health Center in January. Attendance 51. Attendance ratio 0.36.

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WISCONSIN — March 17, 1947. Subject: The Outlook for the Heating Industry and the Relation of the Oil Burner to That Industry. Speaker: W. A. Matheson, director, Eureka Williams Corp., and manager of its Oil-O-Matic Div. Mr. Matheson traced the history of oil burners, indicating a total of 125,000 units being installed in the United States up to the year 1920 and reaching 3,000,000 units by 1947. He also advised that since the nation was definitely geared to petroleum economy in war and peace that it was up to oil burner engineers and petroleum distribution engineers to go to work on new and more efficient oil burner designs and also provide a satisfactory national distribution system. A rising vote of thanks was given to Mr. Matheson. Other Features: Pres. E. W. Gifford appointed J. H. Volk, H. F. Brinen and C. H. Amundson on the nominating committee to present a slate of candidates for the 1947-48 season. J. R. Vernon, alternate to the Annual Meeting, reported on the chapter delegates' meeting in the absence of the delegate, F. J. Nunlist, Jr. M. W. Bishop, chairman of the program committee, announced the future programs for the meetings. Attendance 45.

WISCONSIN—February 19, 1947. Subject: The Future of Atomic Energy as an Industrial Tool. Speaker: Dr. Norman Hilberry, associate director, Argonne National Laboratory. This was the third annual joint meeting with the Engineers Society of Milwaukee and was held in the auditorium of the Wisconsin Electric Power Co., Milwaukee. Dr. Hilberry's talk covered some very interesting highlights on the history and development, as well as difficulties, encountered in the recent atomic energy and atomic bomb program. Following this historical review, Dr. Hilberry discussed at some length the possibilities of the use of atomic energy as an industrial heat and power source, but cited definite limitations based on present knowledge of the science. He indicated that much work must be done in relation to the atomic energy to make it useful as an industrial tool. A lively discussion period followed. Attendance 405.

### Student Branch

TEXAS A & M—March 20, 1947. Subject: What Is Expected of a Young Graduate Engineer in the Field of Air Conditioning. Speakers: W. H. Frost, Dallas Air Conditioning Co., and Harold Broadwell, York Corp., Houston, Tex. Mr. Frost's talk was from the viewpoint of a young engineer who recently entered the profession, while Mr. Broadwell spoke from the viewpoint of an engineer who has been in the profession for many years. Both the talks were of great interest to the members of the student branch and the speakers were well received. A discussion followed each talk and the students derived a great deal of information of particular interest to them. Several members from both the North and South Texas Chapters were in attendance. Attendance 56.

TEXAS A & M—February 20, 1947. Presentation of the charter to the student branch by Dean Barlow, School of Engineering. Dean Barlow gave a short but very inspiring speech concerning the history of the student branch and the future possibilities of its development. O. W. Schuhart, chairman of the placement committee, outlined the functions of this committee and requested all members interested in summer jobs to fill out applications. Acting President Kay announced that the student branch has undertaken the job of installing an air conditioning system in the Mechanical Engineers' Senior Lounge, and invited all interested members to assist in the installation. A discussion took place concerning the representation of the Texas A & M Student Branch on the Student Engineers Council. Attendance 54.

## JARED A. HILL PASSES AWAY ON COAST

Jared A. Hill, a member of the ASHVE since 1937, domestic utilization engineer for the Pacific Gas and Electric Co., San Francisco, Calif., passed away on December 12, 1946, according to word received recently.

Mr. Hill was born at Rodman, N. Y., on December 29, 1893, and received his early education at Watertown, N. Y. He was graduated from Clarkson College of Technology, Potsdam, N. Y., with a B.S. degree in mechanical engineering. His first assignment after his graduation was with the Solvay Process Co., Syracuse, N. Y., as testing engineer. After three years he became assistant master mechanic in charge of heat, light, water and power for the Arnold Print Works, North Adams, Mass. In 1923 he went to California, where he joined the Pacific Gas and Electric Co. In

1926, however, he became engineer for Hunter and Hudson, consulting engineers, San Francisco, and in 1928 he was appointed secretary and manager of the *Heating and Piping Contractors Association*, Oakland, Calif., where he assisted contractors in design and installation of certified heating. He returned to the Pacific Gas and Electric Co. in July 1929 as heating and air conditioning engineer, which position he held until 1942, when he was appointed domestic utilization engineer.

J. A. Hill

Mr. Hill since joining the ASHVE took an active part in the activities of the Golden Gate Chapter, serving as its secretary in 1939 and 1940 and as its vice president in 1941. He was also a member of the *Air Conditioning Society of San Francisco.*

Mr. Hill is survived by his widow, Mrs. Elsie M. Hill, 715 Laurel Ave., Burlingame, Calif., to whom the Officers and Council of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS extend their sincerest sympathy.

## LIFE MEMBER DIES AT 87

George I. Cornwall, a Life Member of the Society, died March 7, 1947, at his home after a long illness.

Mr. Cornwall was born on May 12, 1860, in Hartford, Conn., where he later attended public schools.

He had extensive experience in the heating and ventilating fields, and was considered an authority in his profession.

His professional experience included seven years as superintendent of the pattern department of the Pierce Steam Heating Co. and the American Radiator Co.; and 12 years as superintendent with the U. S. Radiator and Boiler Co., West Newton, Pa. He also served with the Hitchings & Co., Elizabeth, N. J., first as a superintendent and then as manager of their boiler department. For many years he was connected with the Burnham Boiler Corp., as consulting engineer, in which capacity he traveled extensively.

Mr. Cornwall joined the Society in 1919, and was elected to Life Membership in 1934. He was recognized as an authority on greenhouse heating practice.

The Officers and Council of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS extend their sincerest sympathy to his family and many friends.

## DEATH OF P. F. SCHLICK ANNOUNCED

Paul F. Schlick, vice president, Walter Butler Co., St. Paul, Minn., passed away February 19, 1947, after a six months illness.

Mr. Schlick was born in St. Paul, on January 18, 1896, where he received his preliminary education. He attended school in Hackensack, N. J., prior to attending Yale University where he received a Ph. B. in mechanical engineering in 1917.

During the 1st World War he served in the U. S. Army as 2nd Lieutenant in the Infantry. After the armistice in 1919, he secured a position in the Automotive Engineering department of the Minneapolis Steel and Machinery Co. In 1923, he left this company to become vice president of the American Supply Co., also in St. Paul.

Mr. Schlick remained with the American Supply Co., until 1926 when he became manager of the St. Paul office of the Grinnell Co., in which capacity he served until the advent of the second World War. During World War II he was in command of the 216th Coast Artillery Regiment, which had charge of the anti-aircraft defenses of the San Francisco Bay Region. Upon his return to civilian life in September, 1945, he became vice president of the Walter Butler Co.

Mr. Schlick is survived by his widow, Mrs. Paul F. Schlick, and son, Lawrence, who reside at 1410 Edgecume Road, St. Paul, Minn., and a brother, Raymond Schlick, to whom the Officers and Council extend sincerest sympathy.

## DEATH OF JOHN DEVLIN

Announcement of the death of John Devlin on February 11, 1947, after a heart ailment of several months duration, saddened his host of friends in the Delta Chapter.

Mr. Devlin was born on June 29, 1892, at New Orleans, La., where he was educated in a Jesuit school. He attended Loyola University, receiving a B.A. degree in 1910. Four years later he received a B.S. degree in architecture from Tulane University.

His first position was as an engineering draftsman in 1915. This position was interrupted by the 1st World War in which he served as a 1st Lieutenant with the 312th Engineers, 87th Division, spending about one year in France. In 1919 he started his own practice as an architect. However, in 1924, he established the firm of Devlin & Robinson in Okeechobee, Fla. He became affiliated with the Applied Roofing Co., Montgomery, Ala., as manager in 1929.

Since 1931 until his death he had been in partnership with his brother Thomas Devlin, as a representative of several large manufacturers of heating and ventilating equipment.

John Devlin

Mr. Devlin had been a member of the ASHVE since 1939, as well as being a member of the *American Society of Refrigerating Engineers.*

He is survived by his widow, Mrs. John Devlin, a son, John, Jr., three brothers, Thomas, his associate in business, Daniel J. Devlin, now president of New Orleans Assoc. of Commerce, Philip Devlin, Jacksonville, Fla., and one sister, Mrs. Buckner Chipley, New Orleans, to whom the Officers and Council of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS extend their heartfelt sympathy.

# Candidates for Membership

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When the replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 105 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by May 25, 1947, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

AMMONS, STARON E., Student, Texas A & M College. PROPOSERS: W. E. Long, J. S. Hopper. SECONDERS: A. V. Brewer*, C. M. Simmang*.

BAILEY, GEORGE G., Htg. & Air Cond. Engr., Boynton Cole. PROPOSERS: C. B. Cole, W. J. McKinney. SECONDERS: Leo Sudderth, Jr., W. M. Garrard.

BARBECK, CHESTER A., Sales Engr., Matthews Engineering Co. PROPOSERS: R. R. Matthews, J. A. Ray. SECONDERS: E. T. Gessell, J. D. Poythress.

BODKIN, TAYLOR B., Comm'l. Sales Mgr., B & M Electric Co. PROPOSERS: L. L. Gilliland, George Holmes*. SECONDERS: R. O. Turner*, E. K. Leary*.

BRANDT, SANFORD S., Application Engr., Matthews Engineering Co. PROPOSERS: R. R. Matthews, J. A. Ray. SECONDERS: T. H. Anspacher, J. D. Poythress.

BRENNAN, CHESTER T., Distr. Repr., Titusville Iron Works. PROPOSERS: E. B. Cary, L. E. Slawson. SECONDERS: R. L. Byers, W. R. Moore.

BUSH, RICHARD J., Asst. Chief, Air Cond. Div., Abbott, Merkt & Co. PROPOSERS: T. W. Reynolds, O. F. Quist, Jr. SECONDERS: J. C. Fitz, R. J. Ruth.

CLARK, JOE R., Student, Texas A & M College. PROPOSERS: W. E. Long, J. S. Hopper. SECONDERS: C. M. Simmang*, Carl Files*.

COOPER, KENNETH G., Mech. Engrg. Designer, Richard A. Belsham. PROPOSERS: R. P. Kelley, R. J. Clark. SECONDERS: N. L. Hutcheson*, H. C. Shinn*.

CRAWFORD, DAVID H., Student, Texas A & M College. PROPOSERS: W. E. Long, J. S. Hopper. SECONDERS: R. M. Wingren*, Carl Files*.

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SANFORD, ELLWOOD C., Sales Application Engr., Fred W. Kolb. PROPOSERS: C. F. Boester, F. W. Kolb. SECONDERS: T. J. Janes, J. E. Murray.

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SWEENEY, ALLAN R., Engr., The Huffman-Wolfe Co. PROPOSERS: R. W. Pirrung, A. H. Wolfe. SECONDER: E. L. Long, H. R. Allonier.

SZYMANSKI, STANLEY R., Hd., Engrg. Dept., Heating & Air Conditioning Supply, Inc. PROPOSERS: C. M. Mast, R. C. Cushing. SECONDERS: K. O. Alexander, H. G. Smith.

THRAILKILL, HARRY K., Charge of Htg. Dept., American Radiator & Standard Sanitary Corp. PROPOSERS: H. R. Allonier, J. D. Slemmons. SECONDERS: W. M. Myler, Jr., P. C. Vyff.

VAN NIEUKERKEN, J. M., Consultant. PROPOSERS: C. F. Boester, F. W. Hutchinson. SECONDERS: C. R. Deverall, T. E. Veltfort*.

VON ROSENBERG, PAUL C., Prod. Application Engr., Allegheny Engineering Co. (*Advancement*). PROPOSERS: F. C. McIntosh, J. C. Matchett. SECONDERS: P. A. Edwards, H. A. Beighel.

WAGNER, RICHARD E., Chief Engr., H. J. Swanson Co. PROPOSERS: C. E. Parks, R. S. Farr. SECONDERS: Leo Hungerford, F. L. McDonald.

WILSON, HOWARD F., Student, Michigan State College. PROPOSERS: L. G. Miller, C. H. Pesterfield. SECONDERS: K. E. Robinson, Frank Harbin.

WOOD, FRANKLIN J., Vice-Pres., Wood Motor Co. PROPOSERS: F. E. Grosvold, M. W. McRae. SECONDERS: J. D. Pierce, C. E. Towner.

YARDLEY, JACK E., Student, Purdue University. PROPOSERS: C. F. Warner, C. F. Boester. SECONDERS: W. T. Miller, F. W. Hutchinson.

YOUNG, GLENN R., Student, Texas A & M College. PROPOSERS: W. E. Long, J. S. Hopper. SECONDERS: C. M. Simmang*, A. V. Brewer*.

# *Candidates Elected*

In the past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by **Art. B-III**, *Section 8*, of the By-Laws, the following list of candidates elected:

## Members

BAKER, DONALD L., Engr., Carrier Corp., Syracuse, N. Y. (*Advancement*)

BALDWIN, CHARLES W., Vent. Engr., U. S. Rubber Co., Clay, N. Y. (*Advancement.*)

CLAR, ROBERT, JR., Htg. & Piping Engr., Murray W. Sales & Co., Detroit, Mich. (*Reinstatement & Advancement.*)

FLINN, GEORGE S., Air Cond. Engr., J. J. Nolan & Co., Memphis, Tenn. (*Advancement.*)

GRABER, ERNST, Sales Engr., Minneapolis-Honeywell Regulator Co., New York, N. Y. (*Advancement.*)

GROCOTT, J. F. L., Mech. Engr., Anglo Iranian Oil Co., London, England.

HANFMANN, ALEXANDER M., Factory Planning Engr., Western Electric Co., New York, N. Y.

HANTHORN, WALTER, Engr., Kleenair Furnace Co., Portland, Ore. (*Advancement*)

HARRIGAN, PAUL D., Cons. Engr., Harrigan-Hill, New Haven, Conn.

HART, JAMES E., Partner, Watson & Hart, Greensboro, N. C.

KUMMER, CALVIN J., Engr., Carrier Corp., Chicago, Ill. (*Advancement*)

LANGE, FRED F., Pres., The Mechanical Service Co., Minneapolis, Minn. (*Advancement*)

LYFORD, ROBERT G., Branch Mgr., The Powers Regulator Co., Dallas, Tex. (*Advancement*)

RUSSELL, EDGAR N., Tech. Engr., Rosser & Russell Ltd., London, England.

THOMPSON, WALTER G., Research Engr., Duo-Therm Div., Motor Wheel Corp., Lansing, Mich.

## Associate Members

BARDO, FREDERICK J., JR., Designer, Beman & Candee, Buffalo, N. Y.

BARSHA, NICHOLAS B., Chief Engr., Airtherm Manufacturing Co., St. Louis, Mo.

BILLINGTON, DAN B., Sales Engr., A. L. Vanderhoof, Inc., Cleveland, Ohio.

BLATHERWICK, HOWARD E., Warden King, Ltd., Montreal, Que., Canada.

CAIN, JAMES A., Div. Mgr., A. M. Byers Co., Atlanta, Ga.

ELBOGEN, NICHOLAS, Engr., La Brea Heating & Air Conditioning Co., Inglewood, Calif.

EVANS, WILLIAM W., Sales Engr., The Trane Co., Cleveland, Ohio.

FEDER, NATHAN, Chief Engr., E. Kalisch, Inc., New York, N. Y. (*Advancement*)

FIFE, CHARLES M., Mgr., Electric Home Bureau, West Penn Power Co., Pittsburgh, Pa.

FOREST, LEO, Owner, Godin Sheet Metal Works, Montreal, Que., Canada.

GEISER, RICHARD A., Chief Draftsman, Lincoln Bouillon & Associates, Seattle, Wash.

GILLIS, HARLAN W., JR., Mgr., Air Cond. & Refrig., J. Weingartens, Inc., Houston, Tex.

HOSS, ABE N., Sales Engr., Portland General Electric Co., Portland, Ore.

ISERMAN, MELVIN H., Mgr., Htg. Div., General Corporation, Minneapolis, Minn.

JUDSON, ALBERT L., Chief Engr., Iron Fireman Manufacturing Co., Portland, Ore.

KOPALD, NATHAN, Sales Engr., Utility Fan Corp., Los Angeles, Calif.

LATHROP, WILLIAM G., Sales Mgr., T. F. Byrnes, Inc., Hartford, Conn.

LINIGER, RICHARD B., Sales Engr., Liniger Co., Inc., Marion, Ind.

LYNES, GILBERT K., Chief Draftsman, Warden King, Ltd., Montreal, Que., Canada.

MCCLURE, CHARLES J. R., Sales Engr., The Trane Co., St. Louis, Mo.

MONTGOMERY, E. L., Sales Engr., Sheldons, Ltd., Montreal, Que., Canada.

MURRAY, MICHAEL J., Sales Promotion Engr., John J. Nesbitt, Inc., Dallas, Tex.

O'CONNELL, EDMUND F., Partner, Stone Heating & Ventilating Co., Washington, D. C.

PARK, PAUL S., JR., Head of Engrg. Service Dept., A. M. Byers Co., Pittsburgh, Pa.

PAULEY, WILLIAM N., Chief Engr., Farr Co., Los Angeles, Calif.

PERRY, HAROLD E., Dist. Automotive Engr., Socony-Vacuum Oil Co., Inc., Syracuse, N. Y.

PETERSON, ROBERT W., Designer, J. Donald Kroeker, Portland, Ore.

PRESSEUR, HARRY W., Sales & Service Engr., Payne Furnace Co., Houston, Tex.

QUINN, JOSEPH J. JR., Engr., Quinn Bros., Inc., Philadelphia, Pa.

REEVES, RAY O., Purchasing Agent, Airtherm Manufacturing Co., St. Louis, Mo.

SIMMONS, KENNETH A., Mech. Engr., William K. Karsunky, Washington, D. C.

SPANTON, MILTON D., Mgr., Plbg. & Htg. Dept., Coast-to-Coast Stores, Minneapolis, Minn.

SPERRY, HARVEY A., Steam & Hot Water Htg. Inspector, City of Minneapolis, Minneapolis, Minn.

WAGES, ROY W., Div. Indus. Power Engr., Georgia Power Co., Columbus, Ga.

WYBLE, JEROME B., Design Engr. & Draftsman, William K. Karsunky, Washington, D. C.

## Junior Members

BREYER, WILLIAM L., Draftsman & Designer, E. A. Cornely, Inc., San Francisco, Calif.

CARLSON, RICHARD M., Engr., Pfeifer & Shultz, Minneapolis, Minn.

HERMAN, KENNETH R., Purchasing Agent, Armor Insulating Co., Atlanta, Ga.

HESTEKIN, WALTER E., JR., Engr., Hovland Sheet Metal, Eau Claire, Wis.

## Student Members

ASHFORD, FRED, JR., Student, Texas A & M College, College Station, Tex.

BROUSSARD, DOUGLAS E., Student, Texas A & M College, College Station, Tex.

COLLINS, WILLIAM A., JR., Student, Texas A & M College, College Station, Tex.

GUILL, A. W., Student, Texas A & M College, College Station, Tex.

HOYLER, THOMAS H., Student, Texas A & M College, College Station, Tex.

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### C. H. PARKES DIES

Clark H. Parkes, a member of the society since 1944, died in Dodge City, Kans., on March 8, 1947. Mr. Parkes was born on June 27, 1884 at Mt. Vernon, Illinois. He attended public schools in Pratt, Kans., and later enrolled in the University of Wisconsin.

Mr. Parkes represented the Kansas City Pump Co., Kansas City, Mo., from 1917 to 1920, the Chandler Pump & Supply Co., from 1921 to 1932; and the Sterling Engineering Co., Milwaukee, Wis., between 1933 and 1942. In 1943 he became associated with the Trane Co., La Crosse, Kans. At the time of his death Mr. Parkes was sole owner of his own heating, cooling and pumping equipment business at Dodge City, Kans.

Some of the buildings in Kansas for which Mr. Parkes made plans and specifications and superintended the installation of heating plants include Ames Hotel, Syracuse; Grissel Hospital, Ransom; Methodist Church, Cimarron; the Statton Hotel, Mineola; and the Crystal Hotel, Boise City, Okla.

The Officers and Council of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS extend their deepest sympathy to his family in their bereavement.

# AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

## JOURNAL SECTION

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# Officers of Local Chapters

**Atlanta:** Organized, 1937. Headquarters, Atlanta, Ga. Meets, First Monday. President, H. King McCain, 615 Trust Co. of Georgia Bldg. Secretary, Leo Sudderth, Jr., 306 Bona Allen Bldg., Atlanta 3.

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**Central New York:** Organized, 1944. Headquarters, Syracuse, N. Y. President, H. G. Strong, 300 S. Geddes St., Syracuse 1. Secretary, G. F. Keane, 316 Haddonfield Dr., DeWitt, N. Y.

**Central Ohio:** Organized, 1944. Headquarters, Columbus, Ohio. Meets, Third Monday. President, R. B. Breneman, 1282 Edgehill Rd., Columbus 8. Secretary, A. E. Norman, Jr., 1150 Chesapeake Ave., Columbus.

**Cincinnati:** Organized, 1932. Headquarters, Cincinnati, Ohio. Meets, First Tuesday. President, A. W. Edwards, Room 307, 626 Broadway, Cincinnati 2. Secretary, M. E. Mathewson, 1211 Enquirer Bldg., Cincinnati 2.

**Connecticut:** Organized, 1940. Headquarters, New Haven, Conn. President, P. D. Bemis, 36 Pearl St., Hartford 3. Secretary, Winfield Roeder, 405 Temple St., New Haven.

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**Delta:** Organized, 1939. Headquarters, New Orleans, La. Meets, Second Tuesday. President, W. P. Oster, 410 Camp St., New Orleans 12. Secretary, F. G. Burns, 317 Baronne St., New Orleans 9.

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**Golden Gate:** Organized, 1937. Headquarters, San Francisco, Calif. Meets, First Wednesday. President, James Gayner, 624 Hearst Bldg., San Francisco. Secretary, H. V. Hickman, 1129 Folsom St., San Francisco 1.

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**Illinois:** Organized, 1906. Headquarters, Chicago, Ill. Meets, Second Monday. President, Oliver J. Prentice, 450 E. Ohio St., Chicago, 11. Secretary, C. M. Burnam, Jr., Room 1605, 6 N. Michigan Ave., Chicago 2.

**Indiana:** Organized, 1943. Headquarters, Indianapolis, Ind. Meets, Fourth Friday. President, T. R. Davis, 805 K. P. Bldg., Indianapolis 4. Secretary, P. R. Jordan, 311 E. South St., Indianapolis.

**Iowa:** Organized, 1940. Headquarters, Des Moines, Ia. Meets, Second Tuesday. President, E. O. Olson, 3009 Sixth Ave., Des Moines. Secretary, D. C. Murphy, 214 Old Colony Bldg., Des Moines 9.

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**Kansas City:** Organized, 1917. Headquarters, Kansas City, Mo. Meets, First Monday. President, S. L. Furber, 7429 Forest Ave., Kansas City 5. Secretary, H. E. Gould, 1800 Baltimore, Kansas City 8.

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**Manitoba:** Organized, 1935. Headquarters, Winnipeg, Man., Canada. Meets, Third Thursday. President, Robert E. Moore, 107-216 Graham Ave., Winnipeg. Secretary, J. R. Stephenson, 278 Main St., Winnipeg.

**Massachusetts:** Organized, 1912. Headquarters, Boston, Mass. Meets, Third Tuesday. President, P. A. Croney, 72 Arlington St., Newton. Secretary, R. T. Kern, 51 Clafin St., Leominster.

**Memphis:** Organized, 1944. Headquarters, Memphis, Tenn. Meets, First Monday. President, R. H. Hoshall, 65 McCall Pl., Memphis. Secretary, A. T. Bevil, 1521 Waverly Ave., Memphis 6.

**Michigan:** Organized, 1916. Headquarters, Detroit, Mich. Meets, First Monday after 10th of month. President, F. R. Bishop, 8011 Dexter Blvd., Detroit 6. Secretary, L. A. Burch, 5853 Hamilton Ave., Detroit 2.

**Minnesota:** Organized, 1918. Headquarters, Minneapolis, Minn. Meets, First Monday. President, J. E. Haines, 2747 Fourth Ave. S., Minneapolis 8. Secretary, O. L. Lilja, 5000 16th Ave. S., Minneapolis 7.

**Montreal:** Organized, 1936. Headquarters, Montreal, Que., Canada. Meets, Third Monday. President, T. H. Worthington, 405 Beaubien St. W., Montreal, Que. Secretary, S. W. Salter, 910 New Birks Bldg., Montreal.

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**Nebraska:** Organized, 1940. Headquarters, Omaha, Neb. Meets, Second Tuesday. President, H. Y. Kleinkauf, 514 S. 11th St., Omaha 8. Secretary, J. G. Bierl, 4927 N. 34th Ave., Omaha.

**New York:** Organized, 1911. Headquarters, New York, N. Y. Meets, Third Monday. President, W. A. Sherbrooke, 114 Liberty St., New York 6. Secretary, Carl H. Flink, Room 3000, 51 Madison Ave., New York 10.

**North Carolina:** Organized, 1939. Headquarters, Durham, N. C. Meets, Quarterly. President, C. Z. Adams, 312 Piedmont Bldg., P. O. Box 1356, Greensboro. Secretary, M. F. DuChateau, 205 S. Lyndon St., Greensboro.

**North Texas:** Organized, 1938. Headquarters, Dallas, Tex. Meets, Third Monday. President, C. R. Gardner, 1000 St. Louis St., Dallas 1. Secretary, G. A. Linskie, 3124 Milton St., Dallas 7.

**Northern Ohio:** Organized, 1916. Headquarters, Cleveland, Ohio. Meets, Second Monday. President, J. E. Wilhelm, 1906 Euclid Ave., Cleveland 15. Secretary, D. E. Mannen, Jr., 9104 Woodland Ave., Cleveland 3.

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**Oklahoma:** Organized, 1935. Headquarters, Oklahoma City, Okla. Meets, Second Monday. President, G. T. Donceel, Oklahoma Natural Gas Co., Oklahoma City. Secretary, W. W. Frankfurt, 519 Apco Tower, Oklahoma City.

**Ontario:** Organized, 1922. Headquarters, Toronto, Ont., Canada. Meets, First Monday. President, V. J. Jenkinson, 1221 Bay St., Toronto 5. Secretary, H. R. Roth, 57 Bloor St. W., Toronto, Ont.

**Oregon:** Organized, 1939. Headquarters, Portland, Ore. Meets Thursday after First Tuesday. President, C. W. Brissenden, Electric Bldg., Portland 5. Secretary, L. G. Williams, Box 301, Gladstone.

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**Pacific Northwest:** Organized, 1928. Headquarters, Seattle, Wash. Meets, Second Tuesday. President, R. E. Le Riche, 6347 39th S. W., Seattle 6. Secretary, C. W. Finn, 320 8th Ave. N., Seattle 9.

**Philadelphia:** Organized, 1916. Headquarters, Philadelphia, Pa. Meets, Second Thursday. President, E. H. Dafter, 12 S. 12th St., Philadelphia 7. Secretary, L. A. Childs, 330 Harrison Ave., Glenside, Pa.

**Pittsburgh:** Organized, 1919. Headquarters, Pittsburgh, Pa. Meets, Second Monday. President, A. F. Nass, 527 First Ave., Pittsburgh 19. Secretary, E. H. Riesmeyer, Jr., 231-33 Wade St., Pittsburgh 22.

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**Rocky Mountain:** Organized, 1944. Headquarters, Denver, Colo. Meets, First Wednesday. President, J. F. Mohan, 2229 Blake St., Denver 2. Secretary, H. C. Kugeler, 2365 Ash St., Denver 7.

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**St. Louis:** Organized, 1918. Headquarters, St. Louis, Mo. Meets, First Tuesday. President, B. C. Simons, 4620 Chouteau, St. Louis 10. Secretary, W. A. Russell, 7918 Kingsbury Blvd., Clayton 5, Mo.

**South Texas:** Organized, 1938. Headquarters, Houston, Tex. Meets, Third Friday. President, B. P. Fisher, P. O. Box 8, Houston 1. Secretary, C. C. Quin, 809 Stuart Ave., Houston 6.

**Southern California:** Organized, 1930. Headquarters, Los Angeles, Calif. Meets, Second Wednesday. President, Art Theobald, 1161½ S. King Rd. Secretary, R. S. Farr, 2615 Southwest Dr., Los Angeles 43.

**Southwest Texas:** Organized, 1946. Headquarters, San Antonio, Tex. President, F. C. Benham, Jr., 702 S. Flores St., San Antonio. Secretary, L. S. Fawkett, 810 Insurance Bldg., San Antonio 5, Tex.

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**Utah:** Organized, 1944. Headquarters, Salt Lake City, Utah. Meets, First Wednesday. President, E. V. Gritton, 2470 S. 13th E., Salt Lake City 5. Secretary, C. E. Ferguson, 838 Garfield Ave., Salt Lake City.

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**Virginia:** Organized, 1946. Headquarters, Norfolk, Va. President, R. C. Thomas, 819 Westover Ave., Norfolk 7. Secretary, J. F. Reynolds, Gatling and Marble Ave., Norfolk 2.

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**Washington, D. C.:** Organized, 1935. Headquarters, Washington, D. C. Meets, Second Wednesday. President, L. B. Nye, Jr., 1346 H St., N. W., Washington. Secretary, J. N. Stewart, 6124 32nd Pl., N. W., Washington 15.

**Western Michigan:** Organized, 1931. Headquarters, Grand Rapids, Mich. Meets, Second Monday. President, J. W. Miller, Route 5, Box 498, Lansing. Secretary, K. E. Robinson, 211 Smith Ave., Lansing 10.

**Western New York:** Organized, 1919. Headquarters, Buffalo, N. Y. Meets, Second Monday. President, Herman Seelbach, Jr., 45 Allen St., Buffalo 2. Secretary, F. J. Weber, 443 Delaware Ave., Buffalo 4.

**Wisconsin:** Organized, 1922. Headquarters, Milwaukee, Wis. Meets, Third Monday. President, E. W. Gifford, 611 N. Broadway, Milwaukee 2. Secretary, E. M. Kluge, 1817 S. 66th St., Milwaukee 14.

Student Branch

**Texas A. & M. College:** Organized, 1946. Headquarters, College Station, Tex. Meets, Third Thursday. President, Herbert M. Kay, Texas A. & M. College, Box 2336. Secretary, G. H. Jackson, Texas A. & M. College, Box 2737.

# EQUIPMENT DEVELOPMENTS

*For your convenience in obtaining more information about any of this equipment, see coupon on this page. Add the new products and companies listed here to your Directory Section which you received in your January 1947 Heating, Piping & Air Conditioning and thus keep your records of sources of supply up to date throughout the year. Single asterisk indicates equipment not listed in Directory Section; double asterisk equipment and manufacturer not listed.*

## Demineralizes Tap Water for Industrial Use

No. 3441**—A "Filt-R-Stil" water demineralizer, known as the Model U-60, has been developed by the Ion Exchange Products Dept., American Cyanamid Co., 30 Rockefeller Plaza, New York 20, N. Y. The unit is designed to transform tap water into the chemical equivalent of distilled water at a fraction of the costs of distilling, and will handle 60 gph.

Requiring no heat, steam, or auxiliary equipment and having a power consumption of only 6 watts, the demineralizer is designed to deliver water containing a maximum of 10 ppm of ionized solids, or a minimum of 1 ppm. Four columns containing alternate beds of "Ionac" cation and anion exchange resins constitute the bulk of the unit. Water is passed through the columns in series; the cation resin transforms the dissolved salts present in the raw water into acids, and the anion resin is used to absorb the acids.

The unit can also be used to deliver water free of $CO_2$ and is offered for use wherever water is a raw material or a processing medium.

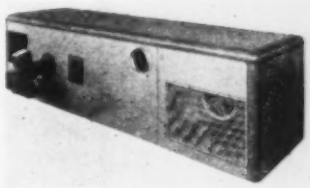
## Expanded Aluminum Foil Used in Grease Filter

No. 3442—A new grease filter, developed by the Research Products Corp., 1015 E. Washington Ave., Madison 3, Wis., utilizes a slit and expanded aluminum foil filter media. Installed in an exhaust canopy over the kitchen range, the unit is said to be 98 per cent effective in catching and holding grease.

The many baffle surfaces of the filter media are designed to prevent grease, dust, lint, and vapors from entering the exhaust system, thus protecting motors and blowers and reducing fire hazards.

Standard size steel holding frames are used, and cleaning is accomplished with hot, soapy water.

## Suspended Unit Heater Is Oil-Fired



No. 3443—The "Shafco" suspended oil-fired unit heater, in single or multiple unit installations, is the solution to many heating problems according to Shafco Distributing Corp., 1354 Book Bldg., Detroit, 26, Mich., sales division of Sawyer Heating Co.

In addition to saving valuable floor space, these overhead automatic units are said to effect marked construction, installation, and maintenance economies. The units are, in effect, self-contained horizontal forced warm air furnaces. They are suspended on steel channel supports from the building wall and roof trusses. Typical examples of buildings suitable for installation of these heaters include gasoline stations, garages, stores, shops, factories, warehouses, sales rooms, public halls, etc., according to the manufacturer.

## San Antonio Has First Air Conditioned City Bus

No. 3444—The first in a fleet of 150 air conditioned city buses has been formally unveiled by the San Antonio Transit Co., and the production schedule called for 50 additional units to be put in service last month, according to an announcement by the Carrier Corp., 302 S. Geddes St., Syracuse 1, N. Y.

The 5 ton air conditioning plant is powered by a separate 16 hp gasoline motor. Cooled air is delivered by circulating fans through a perforated ceiling and through outlets over the seats. To reduce the cooling load, the bodies of the buses are insulated with cork and rock wool, windows are of a heat-absorbing glass, and the sliding window frames may be locked.

In winter the same fans circulate warm air from hot water heaters, located under the floor, through floor ducts to registers beneath the seats.

(*Continued on page* 174)

**Heating, Piping & Air Conditioning** [5-47]

**6 N. Michigan Ave., Chicago 2, Ill.**

**Please ask the manufacturers to send me more information about the equipment mentioned under the following reference numbers in Equipment Developments and Recent Trade Literature.**

***(Circle each number in which you are interested)***

**3441 3442 3443 3444 3445 3446 3447 3448 3449 3450**
**3451 3452 3453 3454 3455 3456 3457 3458 3459 3460**
**3461 3462 3463 3464**
**6497 6498 6499 6500 6501 6502 6503 6504 6505 6506**
**6507 6509 6510 6511 6512 6513 6514 6515 6516**
**6517 6518 6519 6520 6521 6522 6523 6524 6525**

**Name**..... **Title**.....

**Company** .....

**Address** .....

**City**..... **Zone**..... **State**.....

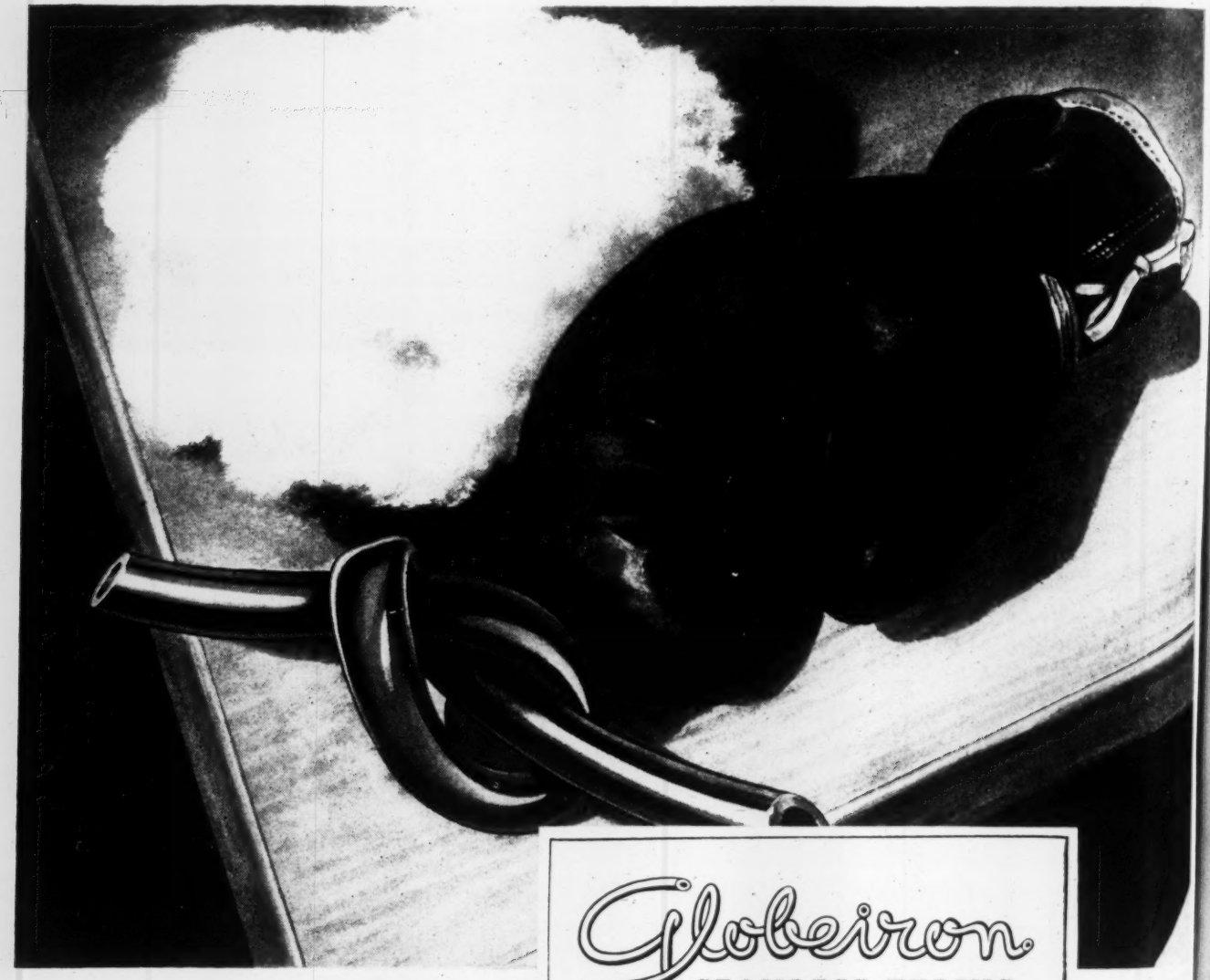
# Soft...tough... ductile

SOFT...tough...ductile — that's Globeiron high purity ingot iron seamless tubing. Tubing that's ideal for severe forming operations—cold or hot. Its metal structure is uniquely uniform — almost pure ferrite. Its magnetic permeability is almost that of pure iron. You can weld it with any common method — no pre-heating or other precautions are necessary.

Globeiron is superior to many steels in corrosion resistance — in any applications wherever corrosive attack is accelerated by segregations and impurities in steels. It may be hot-worked in any temperature range without becoming brittle. Pressure tube requirements, where uniformity, ductility and purity of metal are essential, can best be met with Globeiron seamless tubing.

Globeiron is a product of Globe Steel Tubes Co. — a specialized, dependable manufacturing source for seamless tubing in carbon, alloy, stainless steels, and Gloweld welded stainless steel tubing.

Globe Steel Tubes Co. · Milwaukee 4, Wisconsin



Globeiron
SEAMLESS TUBING

*Write for Globeiron Bulletins Nos. 109A and 113*

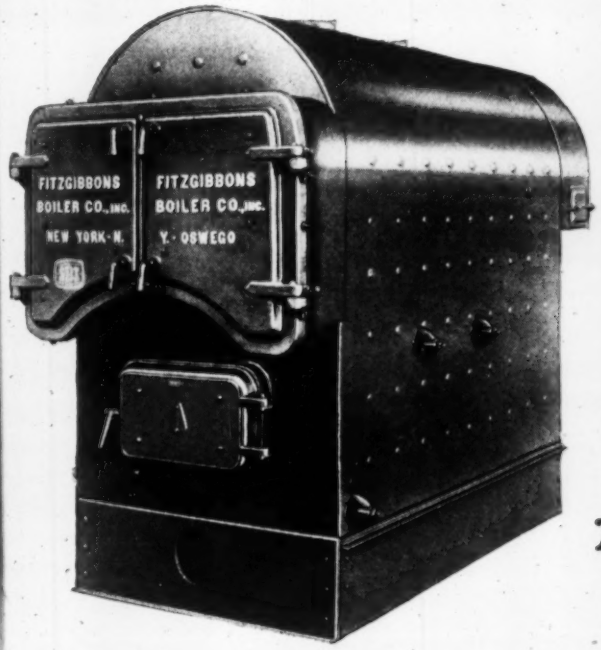
# How to take the gamble out of boiler selection

**HOW CAN YOU BE SURE** you're getting a steel boiler whose construction is in rigid accordance with the standards set by the American Society of Mechanical Engineers?
**Look for the A.S.M.E. stamp on the boiler.**

**HOW CAN YOU BE SURE** your steel boiler has been inspected and hydrostatically tested to insure A.S.M.E. code conformance?
**Look for the "Hartford" inspector's mark on the boiler—your guarantee that he has passed it.**

**HOW CAN YOU BE SURE** the boiler is honestly rated, and will perform right up to or beyond its given capacity?
**Look for the Symbol of the Steel Boiler Institute—on the boiler.**

**LASTLY, HOW CAN YOU BE SURE** that the **design** of your boiler is free of untried and impractical quirks, and is based on sound, proven principles? **Look for the Fitzgibbons trademark, an emblem mark of 61 continuous and successful years of steel boiler building.**




## YOU CAN BE SURE OF *THIS* BOILER

The Fitzgibbons "D" Type—suitable for heating anything from a moderate sized apartment to a large institutional building. On it you will find every one of the "sterling" marks shown above. Built in types for oil, gas, stoker and hand firing, in sizes up to 42,500 sq. ft. steam. "D" Type Catalog, recently published, on request.

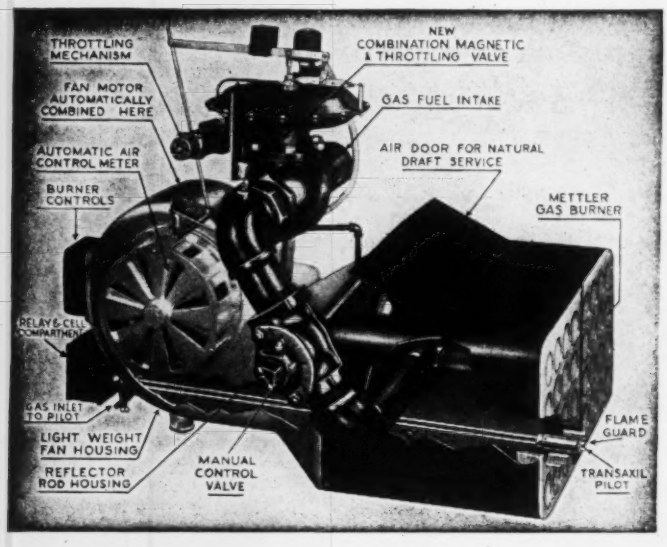
***Fitzgibbons Boiler Company, Inc.***
**101 PARK AVENUE, NEW YORK 17, N. Y.**
Manufactured at: OSWEGO, N. Y.
*Sales Branches in Principal Cities*

# FITZGIBBONS

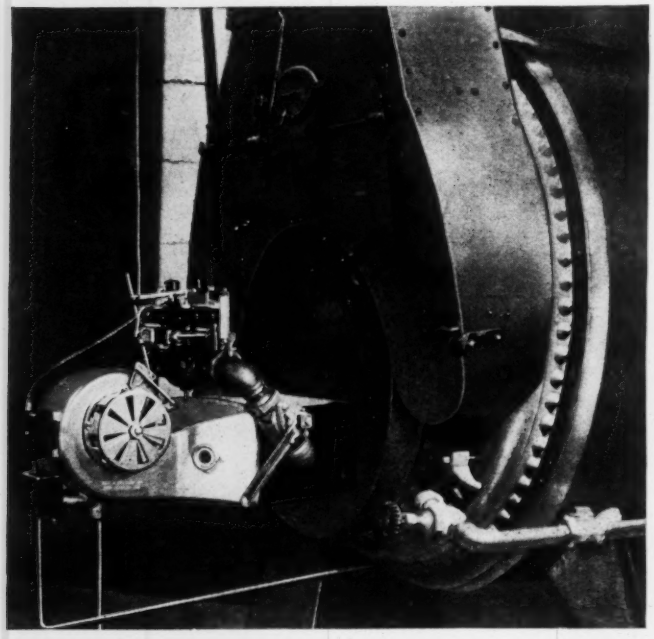
# The Gas Burner

EXPERTO CREDE

# "FAN-AIR"




Picture reveals essential and highly desirable POINTS of "Fan-Air" mech'l draft gas burners. Not just a gas burner but a COMPLETE AND FULLY AUTOMATIC BURNER SYSTEM, factory tested and ready for application when received. It's the most modern gas burner. Why be satisfied with less—when all these tangible advantages are so readily available. It's easier to install, too.



NOTHING ELSE LIKE IT FOR SCOTCH MARINE BOILERS

*Sizes 5 to 300 H.P.* *Prompt Shipments*

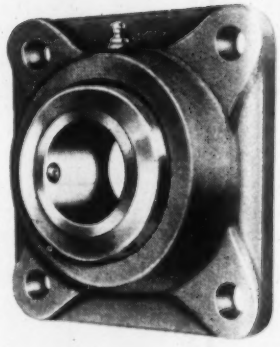
LEE B. METTLER CO.

406 South Main Street

LOS ANGELES 13 CALIFORNIA


(Continued from page 171)

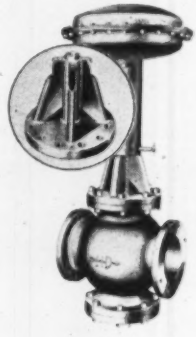
### Ball Bearing Is Self Aligning



**No. 3445—A new type SC ball bearing flange unit has been announced by Dodge Mfg. Corp., Mishawaka, Ind. The bearing is said to be self-aligning and features a long inner race and improved labyrinth seals. It is suitable for mounting against vertical or horizontal frames and supports.**

The back face of the housing is machined and filling slots are also at the back. This is said to provide added protection against external dust and dirt.

### Uses Finned Bonnet on Diaphragm Control Valves



**No. 3446—The Hammel-Dahl Co., 243 Richmond St., Providence 3, R. I., has announced the application of the vertical "U" type fin radiation bonnet for the company's complete line of diaphragm control valves in sizes 2 in. and larger. It is designed to permit natural draft to cool the packing box section of the valve.**

Natural convection currents and radiation are said to dissipate heat and lower packing box temperatures so as to permit the use of the proper type grease seal lubricant, and according to the manufacturer, tests indicate 35 per cent better heat dissipation than with the conventional horizontal fin designs.

The company recommends the use of these radiating bonnets on all control valves when the temperature of the fluid being handled is in excess of 450 F. Extension bonnets are also recommended for handling fluids below the freezing point.

### Inert Gas Shielding Used in Arc Welding Process

**No. 3447—"Heliarc" welding, according to The Linde Air Products Co., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., New York City, is an electric arc welding process which utilizes an inert gas, generally argon, to blanket a single tungsten electrode, the arc, and the weld puddle, and thus exclude the oxidizing atmosphere.**

Because of the protective blanket of inert gas, it is said that aluminum can be successfully fusion welded without the aid of flux, and if recommended procedures are followed, most other metals can also be welded without flux. This, says the company, removes fabrication and design limitations previously encountered.

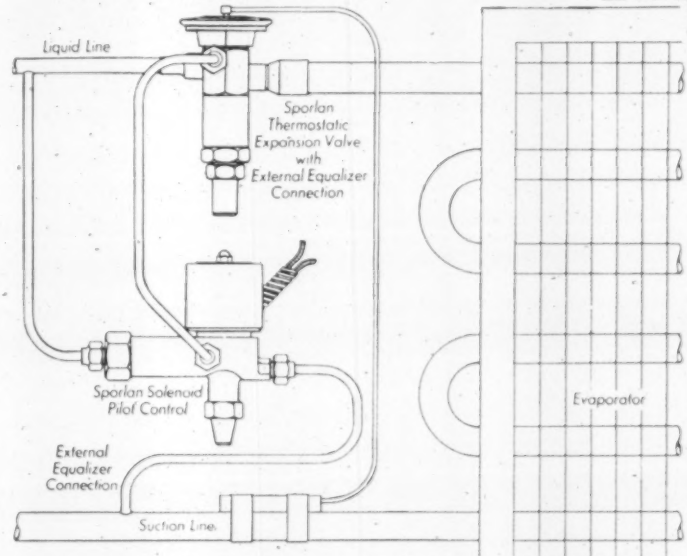
# Save Money 3 Ways

by using the

## SPORLAN SOLENOID PILOT CONTROL

*for Positive Shut-Off of Liquid Lines!*

1. A Sporlan Solenoid Pilot Control costs *less* than a large capacity Solenoid Valve.
2. The Sporlan Solenoid Pilot Control eliminates stocking a variety of large Solenoid Valves. The Sporlan Solenoid Pilot Control is made in only one size and can be used on any capacity job, no matter how big.
3. It's easier and cheaper to install a Sporlan Solenoid Pilot Control than a large Solenoid Valve. The Solenoid Pilot Control is installed in the external equalizer line of a Thermostatic Expansion Valve and only one additional ¼" connection is necessary.




Over-all Dimensions of Solenoid Pilot Control
5½" x 5½" x 3"

## How the Sporlan Solenoid Pilot Control Works...

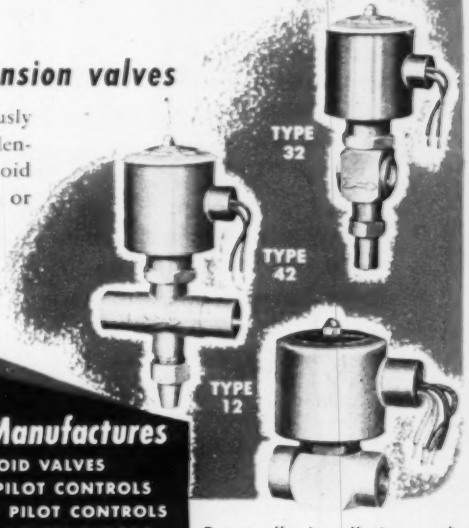
A very small amount of liquid refrigerant is bled from the liquid line through a fine mesh strainer and capillary tube to the equalizer connection. When the Solenoid Pilot Valve is open, the small leak is completely vented to the low side, allowing the true suction pressure to influence the Expansion Valve diaphragm in the usual manner and allowing the Expansion Valve to operate normally at full evaporator capacity. When the Solenoid Pilot Valve closes, liquid pressure builds up under the Expansion Valve diaphragm, overcoming the effect of the bulb pressure and the Expansion Valve spring closes the Expansion Valve tightly.

***Any number and any size of thermostatic expansion valves may be connected to one pilot control,*** thus simultaneously controlling the action of all valves on one evaporator or entire plant. The Solenoid Pilot Control is used in exactly the same manner as a liquid line Solenoid Valve. Its coil is energized either through a thermostat, pressure switch or manual control or by connection across the compressor motor or starter.

### *Two wire control is used . . .*

The expansion valves will be open when the pilot coil is energized and closed when de-energized in exactly the same manner as the conventional solenoid valve. When de-energized the leak from high to low side also stops. The pilot control may be applied to existing jobs merely by connecting with ¼" copper tubing and completing electrical connections.

*Always specify Sporlan when ordering from your wholesaler and get Peak Performance on All installations*



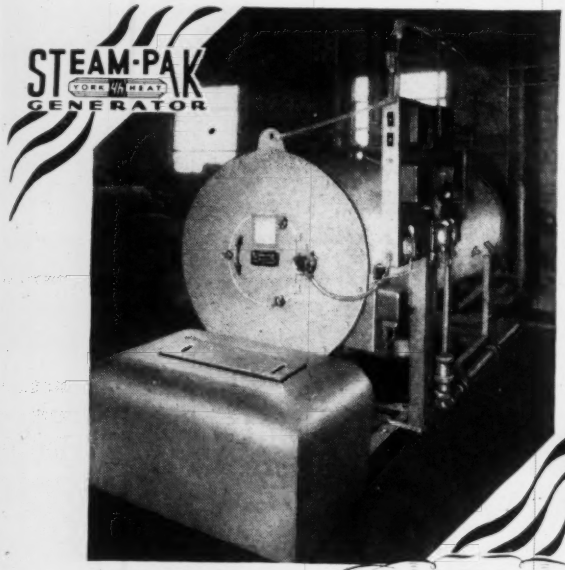

***Sporlan Manufactures***

SOLENOID VALVES
SOLENOID PILOT CONTROLS
MODULATING PILOT CONTROLS
REFRIGERANT DISTRIBUTORS
STRAINERS, CATCH-ALLS
*and the only*
THERMOSTATIC EXPANSION VALVES
*with* SELECTIVE CHARGES

For smaller installations and for ammonia applications Sporlan offers a complete line of conventional Solenoid Valves

**SPORLAN VALVE COMPANY**

3723 COMMONWEALTH AVENUE • ST. LOUIS 17, MISSOURI




Steam-Pak Generator Model SPL-50, installed at Carlisle Baking Co., Carlisle, Pa.

priceless quality
precision combustion

# *Complete* BOILER PLANT IN ONE PACKAGE

Ready to slide into your plant, the Steam-Pak Generator provides boiler, oil burner and accessories including automatic controls; all York-Shipley engineered, to give you greatest operating efficiency and economy. If you need an automatic boiler plant for high or low pressure steam or hot water, the Steam-Pak Generator is the answer.

To insure precision operation of your Steam-Pak Generator, they are distributed *only* through local dealers who are fully qualified to engineer, service and install oil-fired equipment. These dealers are carefully trained in the York-Shipley Policy of "Customer Satisfaction."

Steam-Pak Generators are available in high or low pressure models developing 15, 30, 50, 75, or 100 hp. They are designed for use of light or heavy fuel oils (Nos. 3, 5, or 6). Industrial Division, York-Shipley, Inc., York, Pa.

**YORK-SHIPLEY**

*Oil-Fired Equipment for Industry*

AMERICA'S MOST COMPLETE LINE

IN CANADA—SHIPLEY CO., LTD., TORONTO

Catalogs ID-47-8C and ID-46-1C give complete details on low and high pressure **Steam-Pak Generators.** Yours on written request.


As originally developed during the war by Northrup Aircraft, Inc., helium was the inert gas used; hence the derivation of the process name. The Linde company has been granted an exclusive license for the process.

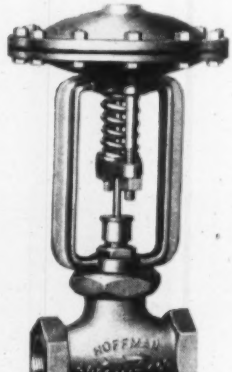
### Air Meter Designed for Low Velocities

**No. 3448*—The Hastings Instrument Co. P. O. Box 1275, Hampton, Va., has introduced an air velocity meter designed to measure accurately air velocities as low as 5 fpm. The unit is applicable to the measurement of velocities in ducts, in air and gas lines, as well as the measurement of wind velocities, and surveys of heating and air conditioning systems.**

The principle of operation is based on a combination of the hot wire and thermopile principles. This arrangement is said to increase the accuracy of the instrument by minimizing the errors due to air temperature variations. The unit operates from 110 volt power and may be adapted for remote indication of velocity or flow of air or other gases. The output is suitable for operating standard strip or circular chart electrical recording instruments.

The standard range is from 0 to 2000 fpm.

### Brass Construction Used in Pressure Reducing Valve



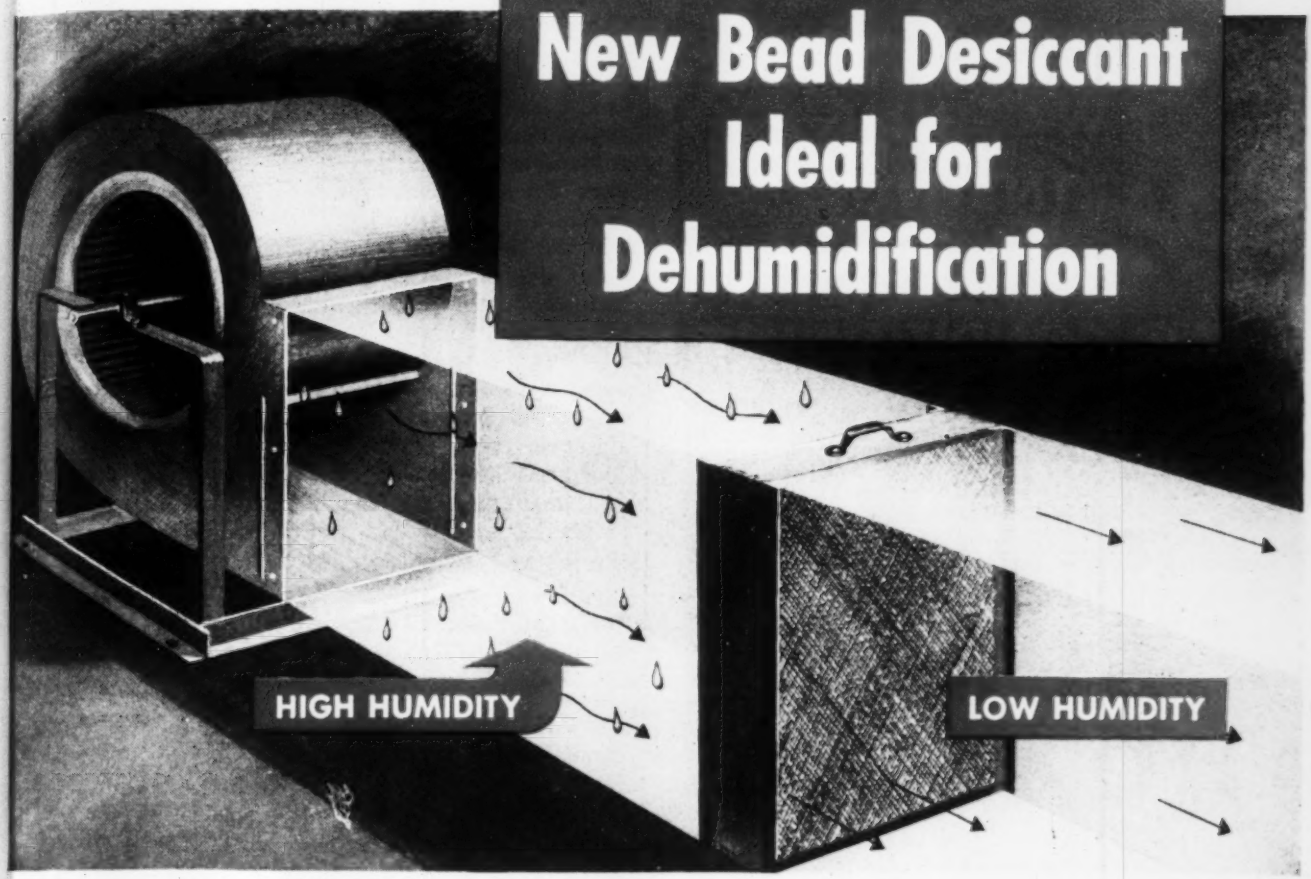

**No. 3449—The new Series 710 pressure reducing valve for either continuous or dead end service has been developed by the Hoffman Specialty Co., 1001 York St., Indianapolis 7, Ind. The maximum high pressure is 250 psi and the low pressure range is from 5 to 80 psi. This single-seated valve is designed for general or universal service.**

Available in sizes from ½ to 1½ in., the valve features a special fabric-inserted rubber diaphragm, deep gland packed stuffing box, and a wide pressure adjustment range. The unit can be used with water or air by a simple change of the disc. For use with fuel oil or gasoline a metal disc is used and the diaphragm is changed from rubber to neoprene or phosphor bronze.

### Sheet Metal Brake Has Special Clamping Action

**No. 3450—A new 24-in. sheet metal brake, "Di-Acro No. 4," has recently been developed by O'Neil-Irwin Mfg. Co., 316 Eighth Ave., S., Minneapolis 15, Minn.**

# New Bead Desiccant Ideal for Dehumidification




## Hard, durable S/V Sovabeads step up efficiency of present and newly designed units . . .

HERE'S NEWS of special interest to both users and manufacturers of dehumidification equipment for air-conditioning. The new bead-type desiccant—S/V Sovabeads—developed by Socony-Vacuum is proving ideal for installation in this equipment.

Honeycombed with ultra-microscopic pores, the small beads suck up moisture like thirsty sponges, provide increased drying capacity and less resistance to airflow than other types of desiccants. Hard and durable, the beads also reduce dusting and attrition losses.

This new desiccant can step up the efficiency of present dehumidification equipment. It has already contributed materially to the design of new type units. One new development, using the beads, features a continuous flow of the desiccant through drying and regeneration processes.

If you are not already familiar with the bead-type desiccant, now's the time to investigate its possibilities for your equipment. Get full details from your Socony-Vacuum Process Products Representative.

SOCONY-VACUUM OIL COMPANY, INC.
26 Broadway, New York 4, N. Y., and Affiliates:
Magnolia Petroleum Co., General Petroleum Corp

Socony-Vacuum Process Products

Process Products

NE IN THE MOBILGAS PROGRAM
DAY EVENINGS, 9:30 E. D. T.—NBC

Gentlemen: Please send me additional information about your new bead-type desiccant, S/V Sovabeads.

Name .....

Company .....

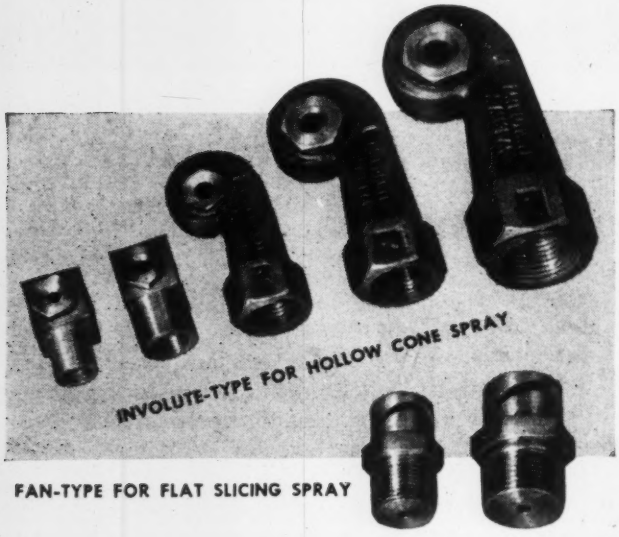
Address ..... State .....

Mail coupon to nearest Socony-Vacuum office or Process Products Division, Socony-Vacuum Oil Co., Inc., 26 Broadway, New York 4, N. Y.

for • SPRAYING
• WASHING
• RINSING
• COOLING
• AIR CONDITIONING

YARWAY

SPRAY NOZZLES




FAN-TYPE FOR FLAT SLICING SPRAY

Get the most out of your Spraying Equipment with minimum power . . . with efficient spraying.

Use Yarway Nozzles. No internal vanes or other restrictions to clog or hinder flow. Two types—Yarway Involute-type producing a fine hollow spray with minimum energy loss, and Yarway Fan-type producing a flat fan-shaped spray with time-saving slicing action for cleaning.

Wide range of standard sizes and capacities. Cast or machined from solid bar stock.

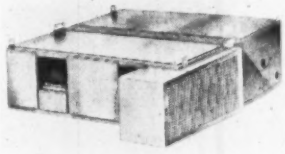
Thousands in use. Write for new Bulletin N-616.

**YARNALL-WARING COMPANY**
**107 Mermaid Avenue** **Philadelphia 18, Pa.**

Unusual features of this unit are said to include special material clamping action which makes possible very sharp bends, the use of roller bearings, an adjustable material gage for precision in duplicating parts and the use of angular degree stops. The unit, weighing 285 lb and incorporating a cast iron base, is designed to accurately duplicate parts in variety of ductile materials such as copper, bronze, stainless steel, aluminum, bimetals, and dielectrics. The brake is capable of handling sheet metal up to and including 16 gage and is arranged so that the original contact surfaces can be readily changed on the job to accommodate the forming of special parts.

### Air Conditioners Feature Installation Flexibility

**No. 3451—Series AVY and AHY central plant air conditioners are designed to permit flexibility of installation, according to Worthington Pump and Machinery Corp., Harrison, N. J.**

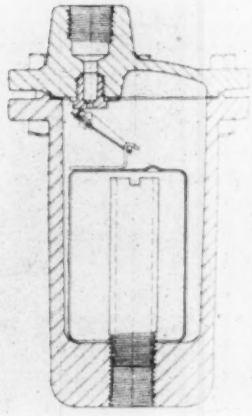


Units for year-round use are available to handle from 1600 to 11,000 cfm. Built in five standard sizes for either direct expansion of "Freon-12" or chilled water cooling, the conditioners may be equipped with steam coils and humidifiers for heating. Electric or pneumatic by-pass control is also available.

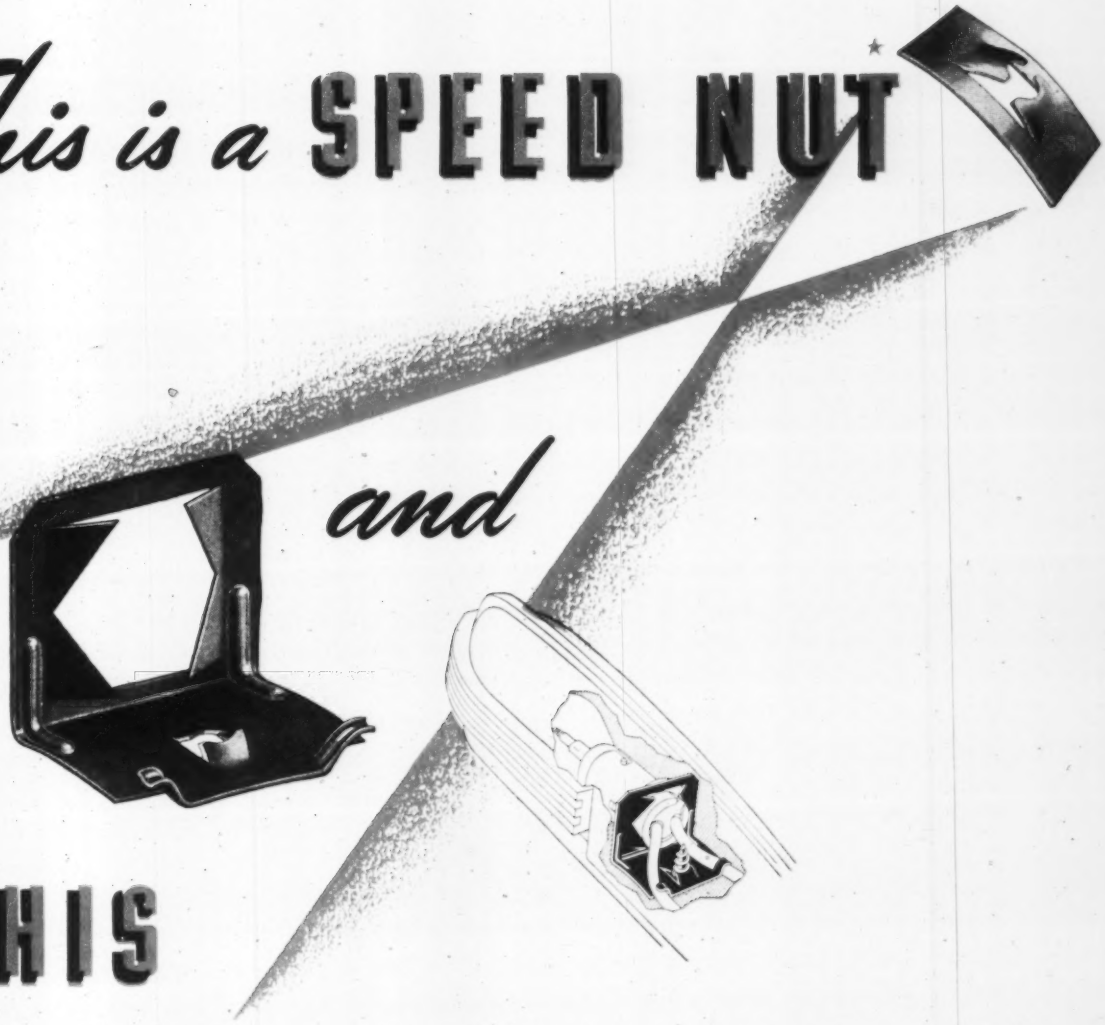
According to the manufacturer, the units may be easily rearranged in the field to accomodate various duct arrangements. Floor mounted units, for example, may be installed for front, top, or rear discharge.

### Forged Steel Steam Trap Has Stainless Trim

**No. 3452—Development of a new, small size, all steel steam trap for service on pressures up to 250 psi has been announced by Armstrong Machine Works, 874 Maple St., Three Rivers, Mich. As shown in the accompanying illustration, the trap embodies the company's inverted bucket and free floating valve lever designs. The body and cap are steel forgings, designed to resist pressures considerably in excess of normal operating pressures.**



# *This is a* SPEED NUT*



*and*

# THIS

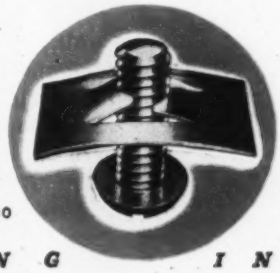
# ..is also a Speed Nut!

Take a look at this one SPEED NUT that does *three* jobs. Conceived by our development engineers to simplify automotive parking lamp assembly, it retains light socket, spaces wires and fastens entire assembly to the base. (See sketch). It replaced four separate parts and eliminated three machining and welding operations.

This is but a typical example of the SPEED NUT Savings Factor. So flexible is the basic SPEED NUT principle, it can be incorporated in almost any conceivable shape to improve your product, speed up your assembly, reduce the number of parts and cut costs. We can give you a no-charge fastening analysis that may surprise you. Send in your assembly details today.

**TINNERMAN PRODUCTS, INC.,**
**2018 FULTON ROAD • CLEVELAND, OHIO**

*In Canada:* Wallace Barnes Co., Ltd., Hamilton, Ontario
*In England:* Simmonds Aerocessories, Ltd., London
*In France:* Aerocessoires Simmonds, S. A., Paris
*In Australia:* Aerocessories, Pty. Ltd., Melbourne



*Speed* PATENTED *Nuts** MORE THAN 4000 SHAPES AND SIZES

* Trade Mark Reg. U. S. Pat. Off.

P A *E S T T H I N G I N F A S T E N I N G S*

WEINMAN Unipumps

Operate efficiently in ANY position . . .

Weinman Unipumps are highly adaptable — they will operate efficiently regardless of the angle of installation on floor or wall. They require minimum space and are easy to install.

Unipumps are especially adaptable for circulation of chilled water or brine. Compactness, efficiency of operation and low maintenance cost, plus a wide range of modifications, make the Weinman Unipump outstanding.

Unipumps are built in sizes to provide capacities of 10 GPM to 1000 GPM, heads to 500 feet, and motor ratings ranging from ½ hp. to 30 hp. The inclosed type impeller is standard but the open type will be furnished if desired.

Type GG Unipump. For detailed information write for Bulletin 726-C2.

Never Forget

• If you need any information concerning pumps or their installation or a letter, telephone call or a telegram will bring an immediate response from our pump specialists.

CENTRIFUGAL SPECIALISTS

WEINMAN PUMP MFG. Company

COLUMBUS • 8 • OHIO

All of the internal operating mechanism, including the valve seat, is stainless steel. The trap measures 4¼ in. in diameter at the cap with an overall height of 7 in. The capacity averages 900 lb of condensate per hour, continuous discharge.

The manufacturer states that this new trap will have special appeal in power plants, steel mills, chemical plants, oil refineries, and other process industry plants, where the trend in recent years has been towards using all steel pipe and fittings for pressure service.

### Magnetic Switching Action Controls Pump

**No. 3453*—A new boiler water level control for providing low water fuel cutoff, low water alarm, and boiler feed pump control by means of a magnetic switching action, has been announced by Magnetrol, Inc., Div. of Fred H. Schaub Engineering Co., Inc., 2110 S. Marshall Blvd., Chicago 23, Ill.**

Changes in water level are transmitted to the switching mechanism by magnetic action, with no mechanical or electrical connection between the pressure chamber and the switch. Model 126, for pressures up to 125 psi, and Model 251, for pressures up to 250 psi, feature the Type S "Unitary" switching mechanism which is self-contained with the mercury switch, magnet, lead wires, and terminal strip built into a single unit.

### Single Pole Mercury Switch Actuated by Snap Tilt

**No. 3454—A new mercury switch, No. AS-4081, designed for almost any switching job, has been developed by the Minneapolis-Honeywell Regulator Co., 2601 4th Ave., S., Minneapolis 8, Minn.**

Less than 1½ in. long, the new switch is rated at one amp for 115 volts, a.c. Simple in design, the device, one of several hundred different models made by the company, is actuated by a snap tilt of 12 deg using a single pole, single throw action.

Production on the new switch has been started and large-scale deliveries are scheduled for later in the spring.

### Room Cooler Built Into Floor Lamp

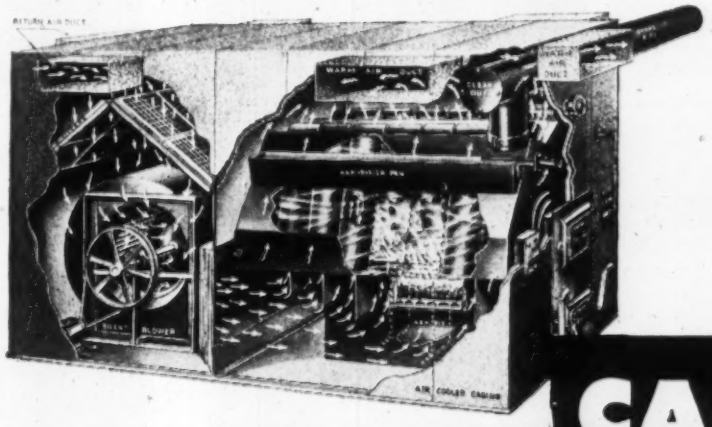
**No. 3455**—A room cooler, utilizing a water-cooled compressor-condenser unit, and a floor lamp have been combined in a unit known as "Luminaire" which is distributed nationally by the Parlong Air Conditioning Corp., 344 Washington Bldg., Washington 5, D. C. Designed for office, store, small factory, and home use, the unit incorporates a fluorescent lighting fixture, and a Westinghouse ultraviolet "Sterilamp" may be added as an optional feature.**

# *Another Efficient Campbell*

# CHURCH HEATING INSTALLATION

*Bergen Luthern Church, Roland, Iowa. This 21-room building is heated comfortably and economically with a single Campbell Heating Unit.*

Campbell 8000 Series Heaters offer four important advantages which make them especially adaptable for heating churches, schools, theaters and other large buildings. They provide fast heat delivery to meet varying "on and off" requirements. They supply abundant heat with remarkable fuel economy. They offer positive, adjustable temperature control; and they are easily adaptable for year 'round air conditioning.



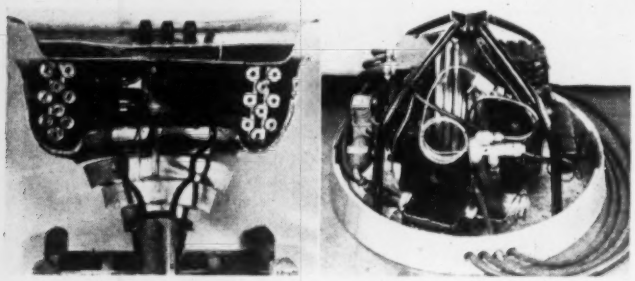
Campbell Heaters a[illegible] efficient, easy to install. Available for all fuels in capacities up to 2,160,000 B.T.U.'s per hour. You are invited to write to our Engineering Department for further information.

**CAMPBELL**
**8000 SERIES**
**HEATERS**

## *Campbell* HEATING COMPANY

DES MOINES 17, IOWA

The insulated dome contains air inlet and outlet louvers, a 1/50 hp circulating fan, and the cooling coil. The 23 in. diameter insulated base assembly houses the hermetic condensing unit, the ½ hp motor, and the necessary controls for automatic operation.



The unit is 69 in. high, uses "Freon" refrigerant, and is said to cool up to 450 sq ft of normal floor area, regardless of ceiling height. According to the company, low velocity air movement increases the air cleaning capacity and eliminates drafts.

### One Piece Fitting Serves Radiators on Hot Water System

**No. 3456**—"Master-Dual-Flow" fittings, offered for use with one pipe, forced hot water heating systems, are designed so that one fitting is used for each radiator. The manufacturer is Mel Products Co., 554 Chapel St., New Haven, Conn.**

Available in either brass or iron, the fitting is inserted in the main and connected to both the supply and return lines for the radiator. According to the company, the unit is foolproof, may be installed in any position, and may be used for any radiator location.

### Cooling Tower Uses Natural Draft

**No. 3457—Designed for both open and closed cooling systems, the new Series 200 natural draft cooling tower may be purchased in various sizes and added to as future expansion requires. The manufacturer is Marley Co., Inc., Kansas City 15, Mo.**

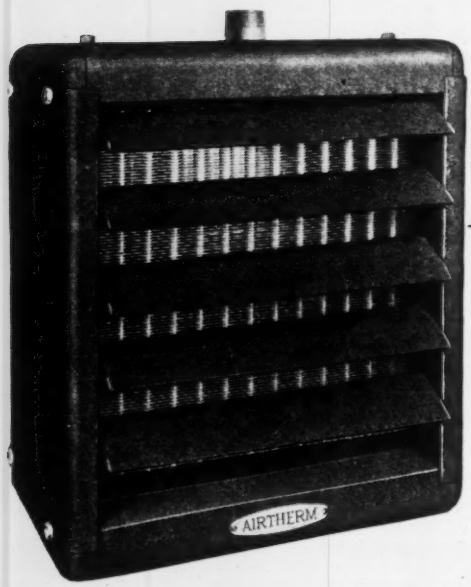
Features of the new unit are said to include standardization for volume production, the ease with which standard atmospheric sections may be installed at any time for closed system cooling, and the use of rough, heavy select heartwood lumber throughout with galvanized bolts and special hardware at critical joints. Capacities range from 65 to 1580 gpm of water.

The company's spray nozzles, said to assure uniform distribution and large volume at low pressures, are standard equipment on these units which are shop fabricated with each piece marked and identified for rapid erection.

### Air Eliminator Valve Designed for Industrial Use

**No. 3458—Development of a new industrial use air eliminator valve for special applications in steam processing operations has been announced by the Lodge & Shipley Co., Special Products Risselt Div., Cincinnati, Ohio.**


**Individually Controlled Airtherm Propeller Fan Unit Heaters Will Cut Your Fuel Bill**




AIRTHERM Propeller Fan Unit Heaters will reduce your fuel costs.

Heat can be uniformly distributed through the use of AIRTHERM Heaters with individual controls, eliminating dead spots in your plant and increasing your heating efficiency.

There is an AIRTHERM Heater for every heating application. Write for Bulletin 1206 giving complete range of available sizes.

**AIRTHERM**
**MANUFACTURING COMPANY**
711 South Spring Avenue • St. Louis 10, Missouri

Known as the J-7 and designed for pressures up to 100 psi, the valve is said to increase efficiency in canning, rubber, food processing, and chemical industries.

### Automatic Electrodes Designed for Production Welding

**No. 3459—Supplementing the "Airco" brand of manual electrodes, the Air Reduction Sales Co., 60 E. 42nd St., New York 17, N. Y., has introduced a group of automatic arc welding wires and tapes.**

This new line embraces five knurled type wires and five tapes all for flat position operation and all recommended for low cost, uniform, automatic welding. Their applications on the production line are said to range from thin gage sheet metal forms to boilers. They are manufactured in several diameters and are designed to meet the general requirements of high speed production welding.

### Continuous Balancing Action Features Recording Pyrometer

**No. 3460—Among the recent features added to the line of instruments manufactured by the Foxboro Co., Foxboro, Mass., is the new "Dynalog" electronic recording pyrometer.**

This instrument has no galvanometer, no slide wire, no dry cell, and requires no standardizing. Because of its continuous balancing action, the recording pen can traverse the full width of the chart in one second's time. Sensitivity is 1/10 of 1 per cent of the full scale range. For applications of temperature measurement, instruments are available with ranges between 200 F and 2800 F, and for use with thermo-couples or electric resistance bulbs. Recorders and recording controllers are offered. A giant indicating scale, for long-distance reading, is an optional additional feature.

In addition to temperature applications, electronic instruments are also designed for the measurement or control of humidity, force, strain, pH, speed, and many other process variables.

### Printing Machine Speeds Up Production

**No. 3461—Intended for large volume production of cut sheets and roll stock up to 42 in. width, the new Model 91 BW "Volumatic" printer-developer has been announced by the Charles Bruning Co., Inc., 4754 Montrose Ave., Chicago 41, Ill.**

The machine is designed to print and develop all of the company's BW mediums. These include light, regular, or card weight paper prints, either in black or in colors on white backgrounds, black or colored line paper prints on green-tinted backgrounds, transparent paper prints, and cloth or film prints. Maximum printing speed is 30 fpm.

New and exclusive features are said to include an enlarged feed board, a simplified control system, increased light efficiency, a tracing and print separator, and a developer designed to provide uniformly developed prints at all speeds.


For Buildings

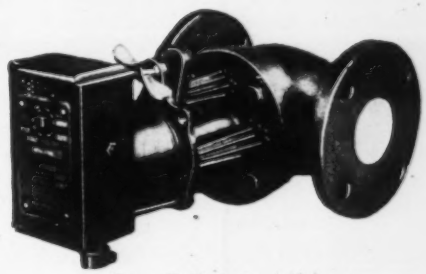
BIG

OR SMALL

TALLMADGE CONTROLS

*Assure Precise Regulation Of Temperatures*

Type A-20 Airotherm



Type C Motorized Valve

Thincup Orifices

***Besides***—Tallmadge Controls offer these money-saving opportunities to the building industry:

(1) Heating Contractors, Engineers and Architects will find Tallmadge Controls are an important factor in figuring lower construction costs on their new projects this summer.

(2) Building Owners can take advantage of this summer's shut-down to modernize out-dated heating systems at reasonable costs, thus reducing fuel and operating expenses for many winters to come.

**What Are Your Problems?**

You are invited to ask Tallmadge engineers how they can lower construction or operating costs for your projects or buildings.

**WEBSTER TALLMADGE & CO., INC.**

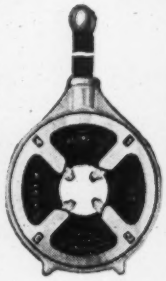
**364 GLENWOOD AVE., EAST ORANGE, N. J.**

QUICK
EASY 1" TO 2"
*Pipe Threading*

RIDGID No. 65R

**10 second adjustment to size instant-setting workholder**

● You can't beat the self-contained semi-automatic 65R for speedy get-ready or smooth perfect threads. Its one set of high-speed steel dies adjust to pipe size in 10 seconds. Mistake-proof workholder sets instantly—one screw to tighten, no bushings to bother with. It cuts clean accurate 1" to 2" threads fast, with least work. Durable all-steel-and-malleable. Save time and effort—buy 65R at your Supply House.

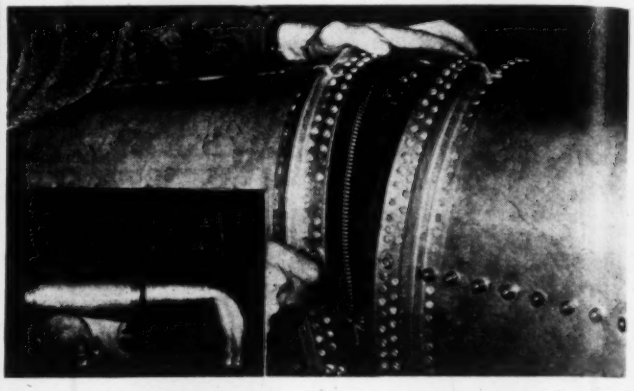


*Every 65R must cut a perfect thread before it leaves the factory.*

RIDGID
WORK-SAVER PIPE TOOLS
THE RIDGE TOOL COMPANY • ELYRIA, OHIO

### Offers Aircraft Pressure Sealing Zipper for Industrial Applications

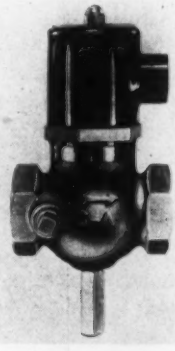
No. 3462—A pressure sealing zipper of rubber and metal, developed by the B. F. Goodrich Co., Akron, Ohio, and called the "Rainbow", is offered for industrial and commercial applications.



The device was originally designed to seal sections of supercharger air ducts on the Republic XF-12, four engine army photographic plane.

### Solenoid Gas Valve Designed for Low Voltage

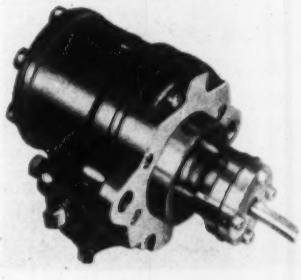
No. 3463*—Gleason-Avery, Inc., 27 Clark St., Auburn, N. Y., has announced production of a new magnetic gas valve, Model SG No. 1. It is designed to control the flow of natural or manufactured gas in heating systems, conversion gas burners, water heaters, and other gas-fired units.



Although it is made essentially for the company's thermostat Model 123-2, it can be used with any two wire, snap action room thermostat. The valve is installed in a vertical position in a horizontal gas line and an arrow on the side of the valve indicates the proper direction of gas flow. The valve should be used with a 115/25 volt transformer.

### Oil Burner Fuel Pump Can Be Quickly Taken Apart

No. 3464—Polo Pump Co., Polo, Illinois, is embarking upon a program of substantially increased production of their Type 12 oil burner fuel pump, rated at 1725 rpm, 100 psi pressure, 12 gph, and having a pressure range of 75-150 psi and a maximum vacuum of 25 in. of mercury.



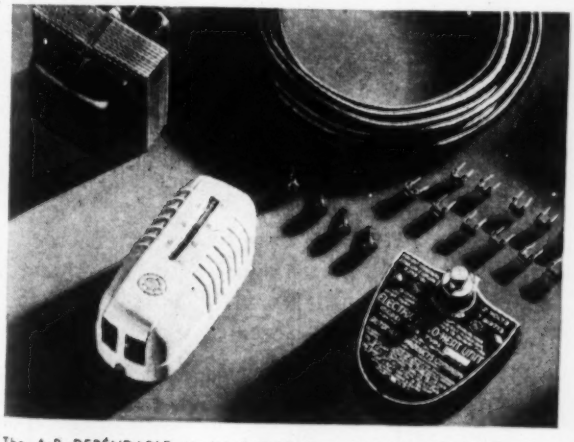
The manufacturer emphasizes the fact that the entire unit can be disassembled with a screw driver

# MAKE *Extra* SUMMER PROFITS

**HELP YOUR OIL HEATER CUSTOMERS TO NEXT WINTER'S FUEL SAVINGS . . . *NOW!***

● You can install THERMOSTATIC Temperature Control on most of the oil heaters you have sold since 1939! That means extra profits on every sale you made during the last 8 years!

Most of your oil heater customers — even those as far back as 1939 — can enjoy the luxury and economy of A-P THERMOSTATIC Temperature Control. And NOW is the time to tell them about it, when their heaters are inactive during the summer months. If their heater uses the A-P Constant Level Oil Control Model 240-DR, UR, or YR you can easily and inexpensively install a new A-P Model 240-ED Heat Regulator Set. They'll thank you next winter, when their fuel bills run 10% to 25% lower, and they're enjoying the pleasure and convenience of steady, uniform heat without the danger and discomfort of overheating — at the mere setting of a *thermostatic dial*.



The A-P DEPENDABLE Model 240-ED Heat Regulator Set. Includes an Electric Conversion Top to mount on present Manual Control, accurate Wall Thermostat, Transformer and all accessories and instructions.

**WRITE TODAY for details and prices — and supply of colorful direct mail folders to help you to ready SUMMER PROFITS on A-P Heat Regulator Sets.**

Be sure your new heater lines use · · · A-P DEPENDABLE OIL CONTROLS — for faster sales, greater customer satisfaction.

**AUTOMATIC PRODUCTS COMPANY**
2460 North Thirty-Second Street, Milwaukee 10, Wisconsin

**DEPENDABLE** *Oil Controls*

**DESIGNED TO ELIMINATE SERVICING**

# A CINCH TO CUT PIPE *Extra Fast*

# RIDGID Pipe Cutter

## is efficiently-balanced for clean speedy work

● This new style RIDGID is easy on your time and muscle; rolls through pipe extra fast, extra easily, leaving practically no burr. Made to toe the mark by its neatly balanced well designed malleable frame, the heavy-duty tool-steel wheel always cuts true. Every cutter is factory-tested. Choice of 5 sizes to 6" capacity, 4-wheel cutters to 4." You'll like this speedy work-saver — buy it at your Supply House.

*4-wheel No. 42 for fast quarter-turn cutting.*

**RIDGID**

**WORK-SAVER PIPE TOOLS**

THE RIDGE TOOL COMPANY · ELYRIA, OHIO

**For Insulation Jobs, too**

# IT HAS TO BE RIGHT THE FIRST TIME

**. . . and the first choice for the right insulating job is the Johns-Manville Insulation Applicator.**

Even the finest insulation must be properly applied to provide maximum efficiency. That's why leading refrigeration engineers have long relied on the combination of Johns-Manville Rock Cork and the skill of J-M Applicators.

Basically mineral in composition, Rock Cork is made in both sheet and pipe-covering form. Both provide advantages that plant operators and engineers appreciate . . . low thermal conductivity, exceptional resistance to rot, moisture and vermin.

And you can be sure that Rock Cork will be applied to give you maximum use of its high insulating efficiency. J-M Insulation Applicators are fully trained in all phases of application. They employ only skilled mechanics and the most up-to-date methods of application.

For further details, write for brochure DS-555, or see your J-M Insulation Applicator. Johns-Manville, Box 290, New York 16, N. Y.

JM PRODUCT

**Johns-Manville ROCK CORK**

and a set screw wrench, and all parts of the valve can be removed with the fingers as there are no press fits. These features are designed to simplify maintenance. The strainer consists of four nested cylinders of fine mesh monel screen. The shaft is hard chrome plated, and has a leakproof, long life mechanical seal. Gear pump members are of special design, shaved and lapped for accuracy and smooth, quiet operation. With adapter flanges, available in three types, units are said to be ideal as replacements for existing burners.

---

### Equipment "Shorts"

"The largest display of its kind ever built," according to the designers, introduced the 1947 products of York-Shipley, Inc., York, Pa., at the recent heating and ventilating exposition. Main feature of the 100 ft display was a lighted iris shutter which opened and closed constantly. The shutter is used in the company's line of industrial and commercial oil-fired heating equipment.

Buffalo Forge Co., of Buffalo, N. Y., and its subsidiary, Buffalo Pumps, Inc., have announced firm prices on all lines of their products. This decision was reached after considerable deliberation with the thought that the country's best interests will be served by a return to normal pricing procedure.

A new technique for measuring the amount of water in "Freon" refrigerants and certain other organic liquids and gases with an accuracy of one part in a million was recently described to the American Chemical Society in a paper by A. F. Benning, A. A. Ebert, and C. F. Irwin of the organic chemicals department, E. I. du Pont de Nemours & Co. They told of research which was carried out at the request of Kinetic Chemicals, Inc., a du Pont affiliate, to further improve the purity of fluorinated hydrocarbons.

The addition of "Wypo" tip cleaners, for use in cleaning oxyacetylene welding and cutting tips, to its line of apparatus has been announced by Air Reduction Sales Co. of New York.....The Trane Co., La Crosse, Wis., manufacturers of heating, cooling, and air conditioning equipment, has announced that all orders received for early delivery will be accepted at firm prices. All escalator clauses have been removed.

An Engineering Reason is the title of a 30 minute color and sound motion picture covering the generation and utilization of steam at low, medium, and high temperature and pressure conditions. Use of the film is being offered to engineering and business association groups and industrial plant personnel by Yarnall-Waring Co. of Philadelphia, manufacturers of steam plant equipment.....A new, self-adhesive "Kum-Kleen" label for carrying the "UL" approval of the Underwriters' Laboratories is offered by Avery Adhesive Label Corp. of Pasadena, Calif. The label is said to stick permanently to any smooth surface, yet is easily peeled off.

A new west coast service plant at 3625 Medford St., Los Angeles 33, Calif., has been opened by Handy

SMART ENOUGH

NOT TO WORK

ON THAT!

A squirrel has a hard time making a living. He can't afford to do useless and unprofitable work.

So, he inspects every nut he picks up. If he gets hold of a bad one, he throws it away.

The plumber or steamfitter doesn't have to inspect "K" fittings. Bad ones are thrown away at the factory. This saves the work and grief which would result from misalignment, leaks, call-backs.

*Insist on "K" fittings.*

- Standard and Extra Heavy Cast-Iron Screwed Fittings
- Standard Flanged Cast Fittings
- Standard and Extra Heavy Companion Fittings
- Drainage Fittings

*Catalog on request*

**KUHNS BROTHERS CO.**
Dayton 1, Ohio

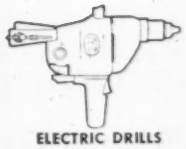
**K** PRECISION FITTINGS

*COMBINED MARKET FACILITIES at Malleable Iron Fittings Co., Branford, Conn., and at Kuhns Bros. Co., Dayton, Ohio.*



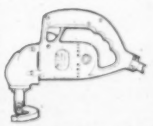
# Hammers Down Costs On Jobs Like This!...

B & D Hammer channeling in concrete floor.

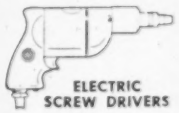


ELECTRIC DRILLS

PORTABLE SANDERS



ELECTRIC SHEARS



ELECTRIC SCREW DRIVERS

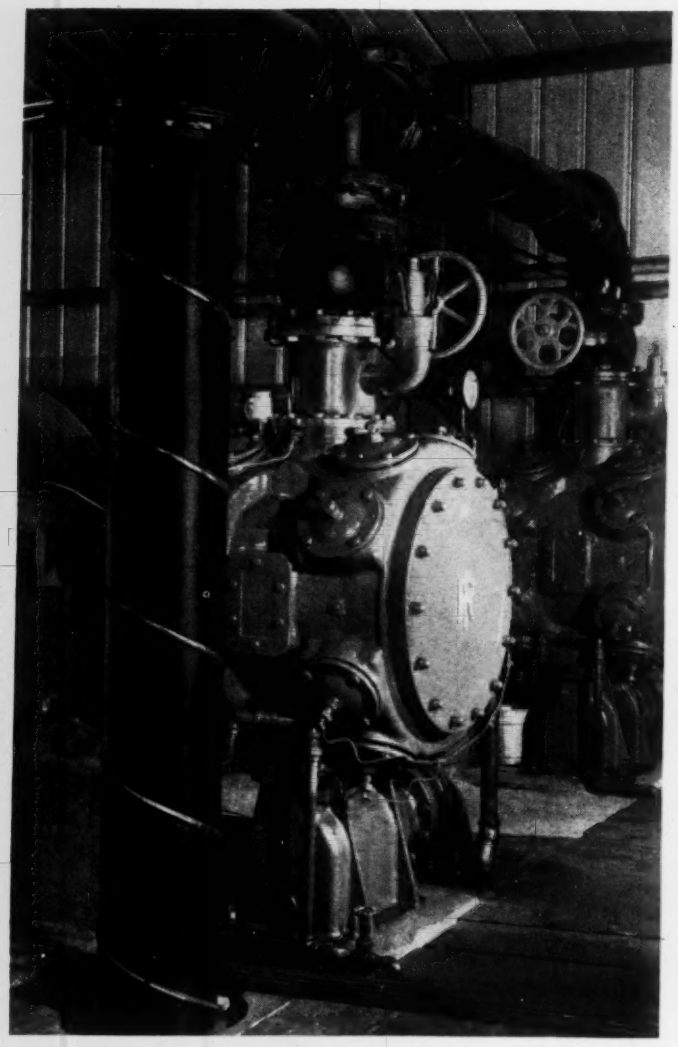
You'll find the Black & Decker Electric Hammer a handy tool for plenty of the installation work you're now doing slowly and laboriously by hand. With this high-speed, hard-hitting tool you can drill holes in concrete, stone, brick or masonry for installing air conditioning ducts and piping . . . channel in concrete flooring to conceal pipes, conduits, etc. . . . handle over-head work a lot more easily.

Ask your nearby Black & Decker Distributor today about this time-saving tool. Black & Decker Hammers are amazingly husky, yet light in weight, and completely self-contained. Operate from A.C. or D.C. outlet or portable generator. Require no transformer or extra equipment. Powered by special, Black & Decker-built Universal motor that delivers thousands of sharp, positive hammer blows per minute. For details on Hammers and many other Portable Electric Tools for heating, piping and air conditioning work, write for our catalog to: The Black & Decker Mfg. Co., 628 Pennsylvania Ave., Towson 4, Md.

LEADING DISTRIBUTORS EVERYWHERE SELL

**Black & Decker**

**PORTABLE ELECTRIC TOOLS**



**NAYLOR PIPE for AIR TRANSPORT**

Where there is air to be moved, today's ideal vehicle is Naylor light weight pipe. More and more, this distinctive pipe is being used for push-pull ventilating service and high or low pressure air lines. The exclusive Lockseam Spiralweld makes Naylor Pipe tight, leakfree, stronger and safer. This structure creates a reinforcing truss which provides additional collapse strength and permits the use of Naylor Pipe on air jobs that no other light weight pipe can handle. The larger the diameter pipe required, the more this is true. Write for the complete story in Catalog No. 44 today.

Naylor

**NAYLOR PIPE COMPANY**

1251 East 92nd Street • Chicago 19, Illinois

New York Office: 350 Madison Avenue • New York 17, N.Y.

**Equipment "Shorts" (Cont.)**

& Harman Co. of New York City. The new plant will house a sales department and manufacturing facilities. All of the company's products including silver brazing alloys and fluxes, sterling silver, gold sheet and wire, and silver anodes will be produced at this plant. . . . . A 16 mm motion picture, Solid Comfort, has been produced for the Celotex Corp., of Chicago. While it is a sales promotion production, it is said to be unusually interesting and educational. The manufacturing and application of the company's insulating materials are featured. The film will be available to groups and organizations in the near future.

The J. H. Williams & Co. of Buffalo, N. Y., has broadened its line of drop-forged tools by the addition of "Flange Jacks," formerly manufactured by the T. G. Persson Co. of Bloomfield, N. J. Used in pairs, the device is designed to simplify gasket replacement.

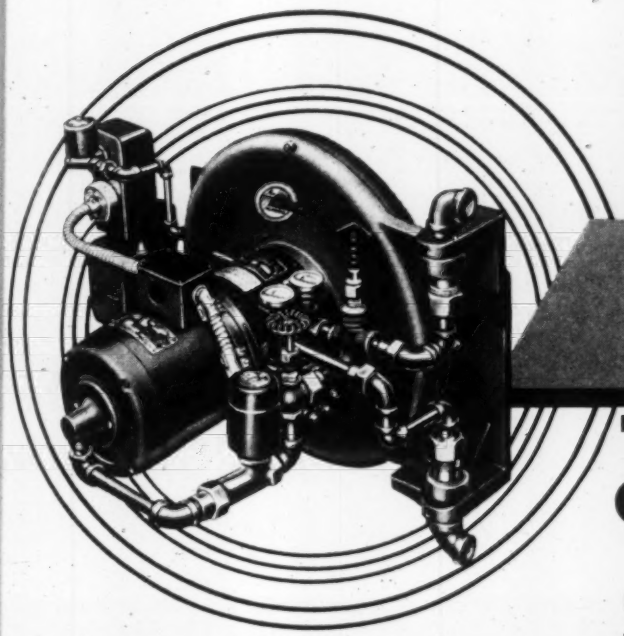
A complete line of stainless steel electrodes in a full range of grades and diameters has been announced by Wilson Welders & Metals Co., Inc., of New York City. All stainless steel electrodes are furnished with an extruded lime type coating for d-c application, and in addition, all but the straight chrome analyses are obtainable with a lime-titania coating which is usable on a-c or d-c.

Production of building insulation has started in the new Kansas City plant of the Owens-Corning Fiberglas Corp. which has also announced the purchase of a new plant site in Santa Clara, Calif. Meeting the needs of west coast builders and contractors for various types of insulation is the immediate purpose of the California unit.

The new Tyler, Tex., plant of the Bryant Heater Co. was the scene of a recent annual company sales conference. New models in the gas heating field which were on display included unit heaters, water heaters, and furnaces . . . . . A recent order received by the Foxboro Co. marked the sale of the 18,000th copy of Principles and Practice of Flow Meter Engineering. This well known handbook, now in its sixth edition, is widely used in many industrial fields.

Minneapolis-Honeywell Regulator Co., has formed a new sales division to market its warm air registers and grilles. Limited production of the new air conditioning register and the new return grille is under way at the company's Wabash plant. Distribution and sales will be handled through branch offices. . . . . The Skinner Chuck Co., New Britain, Conn., has purchased the Allied Control Valve Co., Inc. of South Norwalk, Conn. This company is now being operated as a division of Skinner and will function as a separate plant designing and manufacturing solenoid valves.

"Flexonics", according to an announcement by the Chicago Metal Hose Corp., of Maywood, Ill. is defined as "the science of controlled bending of thin metals for use under varying conditions of temperature, pressure, vibration, and corrosion." The newly coined word is said to have particular reference to the manufacturing of flexible metal hose.



# Ace UNIFLOW

## HORIZONTAL ROTARY OIL BURNERS

**FOR PERFECT HEAVY OIL BURNING UNDER ALL OPERATING CONDITIONS**

*Models Available*

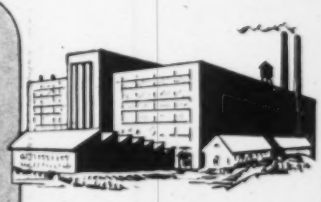
| | GALLONS PER HOUR | BOILER H. P. | SQUARE FEET STEAM RAD. |
|---|---|---|---|
| MINIMUM | 2 | 6 | 800 |
| MAXIMUM | 165 | 500 | 64,000 |

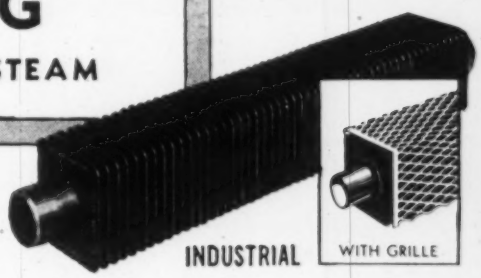
**ACE ENGINEERING CO.**
1435 W. 15th Street • Chicago, Illinois

*Custom Engineered Oil Burning Systems Since 1931*

## *For* INDUSTRIAL *and* RESIDENTIAL HEATING

**WITH HOT WATER OR STEAM**






Vulcan baseboard radiator (Radi-Vector) has offered to the heating trade a radiator for residential or domestic use of the same high quality as the standard Vulcan finned type radiator which has been used by industrial concerns and railroads for over two decades.

In addition to its light weight and space saving, uniformity of heat has made this type of radiation popular with architects, heating contractors, steamfitters and users.

The solid metal front of the baseboard radiator gives radiant heat in addition to the heat by convection from the fins.

**THE VULCAN RADIATOR CO.** 26 FRANCIS AVENUE, HARTFORD 6, CONN.
*RADIATOR MANUFACTURERS FOR OVER TWO DECADES*

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# NEW BOOKS & REPORTS

## Dr. Giesecke Publishes Book on Hot Water Heating, Radiant Heating, Radiant Cooling

A past president of the ASHVE and at present a consulting engineer, Dr. F. E. Giesecke has devoted a lifetime to improving the art of heating with water. As head of the division of engineering research of the University of Texas and as director of the Texas Engineering Experiment Station of the Agricultural and Mechanical College of Texas, he conducted research that determined resistance to flow of water in pipes, particularly within the range of sizes and velocities encountered in forced hot water heating. Much of the information on the mechanics of hot water heating systems presented in his new book therefore is descended lineally from his own research.

This book—titled Hot Water Heating and Radiant Heating and Radiant Cooling—is intended to serve as a summary of all knowledge the author has acquired relating to water as used to heat buildings. It embraces comprehensively the popular subject of heating by radiant means from surfaces warmed by water in pipes and by air in confined paths, and it discusses the newer and less common application of cooling by radiant heat transfer.

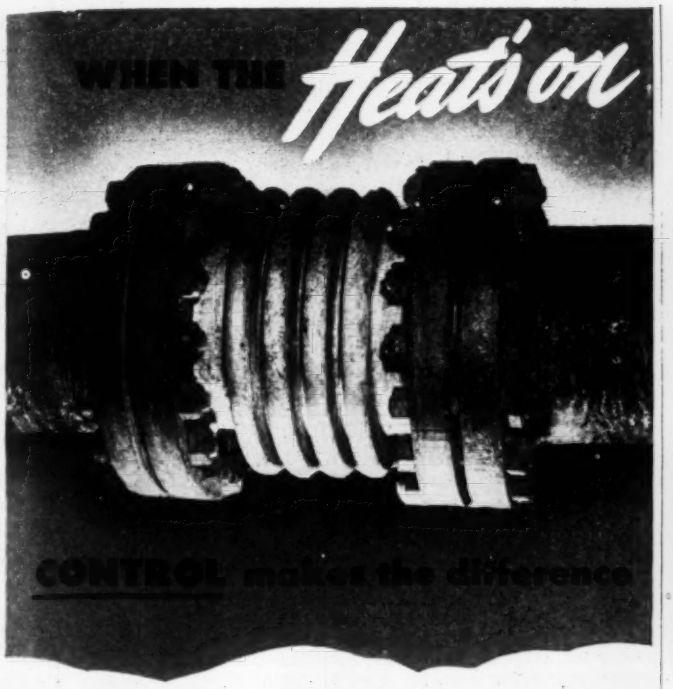
The logical composition of the book could have been conceived only by an experienced teacher. The initial chapters define the many classifications of hot water heating systems, then discuss briefly the heating requirements of buildings and the heat emission of heat transmitting components of heating systems such as radiators, convectors, and panels. These data expressly are supplementary to that of standard reference manuals such as the ASHVE Guide.

Following this preparatory outline, 10 chapters are devoted to design of elementary hot water heating systems. These examples are meticulously detailed and copiously illustrated with drawings, diagrams, facts, and figures. A separate chapter is used to analyze piping systems employing open expansion tanks and those employing closed expansion tanks. Another chapter is devoted to district heating with forced circulation hot water. These chapters likewise are illustrated with drawings, diagrams, and examples.

The explanatory section is followed by four chapters intended to serve as reference outlines for design of four types of heating systems: a one pipe, forced circulation system; a one pipe, gravity system; a two pipe, reverse return, forced circulation system; and a warm water radiant heating system.

Directions are set forth in a logical series of steps which a designer would take in preparing these designs. There are numerous cross references to previous and subsequent paragraphs in which the development of the principle is made.

That part of the book devoted to radiant heating impressed this reviewer as being the most useful condensation of information on the subject that he has seen. Starting with a dissertation on fundamentals, the book explains by means of examples the deter-

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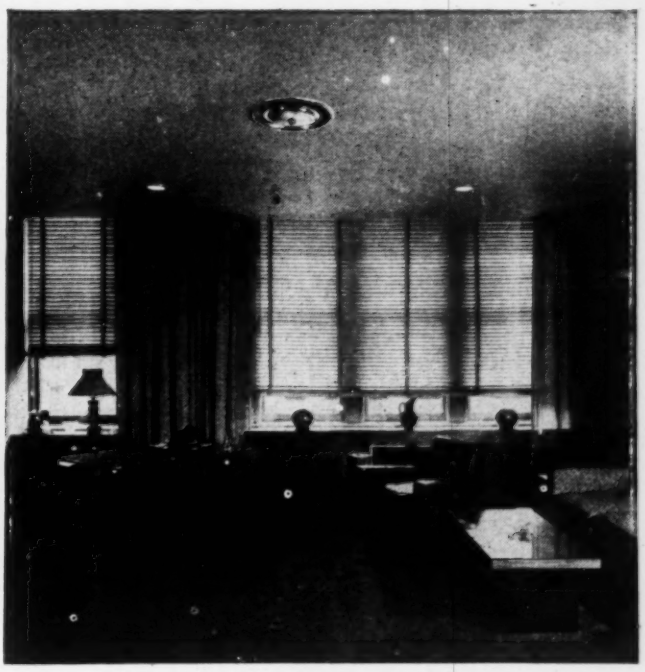

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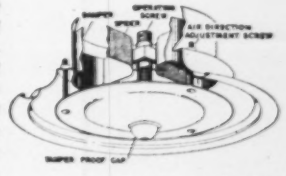


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minations of required surface temperatures for panels. This is followed with three examples of the design of warm air heating panels and five examples of warm water heating panels in floors and ceilings. Several charts and tables are presented to assist the designer.

The author gives frequent empirical values that may be used in the design of heating panels. These are stated to be based on reasonable assumptions, usually following the mathematical approximation of the values. It is recalled that long ago Dr. Giesecke advanced the value of 3.5 Btu per hr per sq ft pipe surface per deg temperature difference water to air as the rate of heat flow for usual warm water heated panels. This was based on his own research and practical experience. It has been substantiated by several successful radiant heating installations designed in the reviewer's office.

Conspicuous by its absence, however, is any reference to use of copper pipe for warming panels with water. All data presented for design of warm water heating panels appear to be based on use of black iron pipe.

The author makes no attempt to guide the designer in selecting the wall or ceiling or floor for the warmed panel location. The text is devoted to the technique of designing warm panel areas *per se*.

In the radiant cooling section of the book the relatively few installations that have been made in Europe are described and discussed. The limitation of use of radiant cooling for sensible heat extraction necessarily restricts its applications to buildings in relatively dry climates and except those in which large numbers of people congregate.

The development of design procedure for cooling panels is made in logical order, similarly to that presented for heating panels.

The radiant heating and cooling section of the book definitely is not a theoretical treatise on the mathematical conception of energy transfer by radiation such as would delight a purist; instead, it is a useful text and guide for the designer, full of practical instruction in the design and construction of radiant heating and cooling systems. In fact, this statement may be extended to apply to the entire book.

This book comprises 260 + ix pages, it is published by the Technical Book Co., P. O. Box 62, Austin, Texas, and its price is $4.00.—*Reviewed by S. I. Rottmayer, of Samuel R. Lewis & Associates, consulting engineers.*

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### Technical Developments to Feature British Exhibition

The British Engineers' Association, 32 Victoria St., London, S.W.I., England, has announced the 1947 Engineering and Marine Exhibition. This show is the first since 1937 in a long series and, according to the association, will "provide an opportunity of seeing all that is of importance in technical development over the past decade." The show will be held August 28 to September 13 in London's largest exhibition building, the Olympia.

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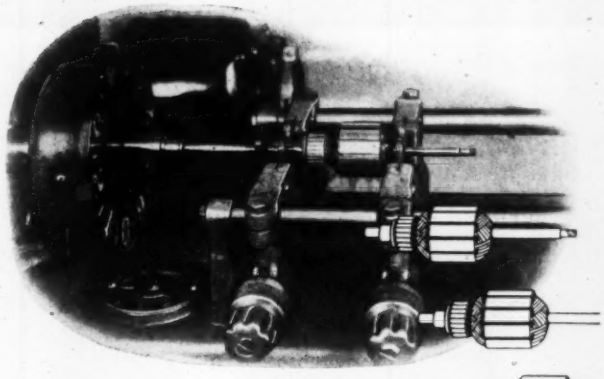

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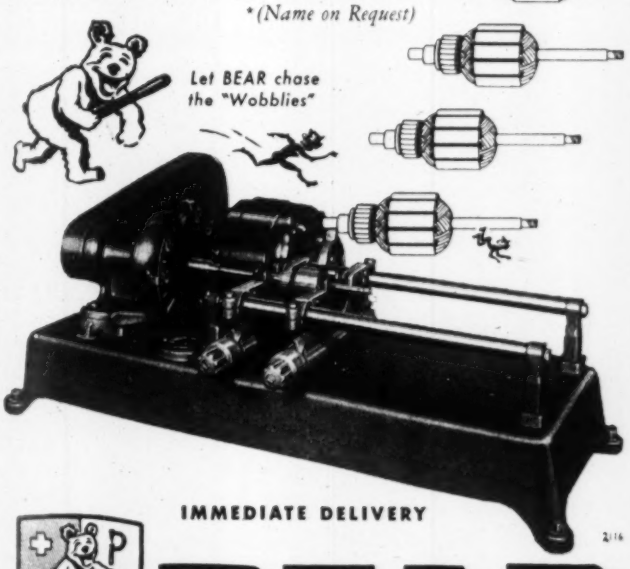


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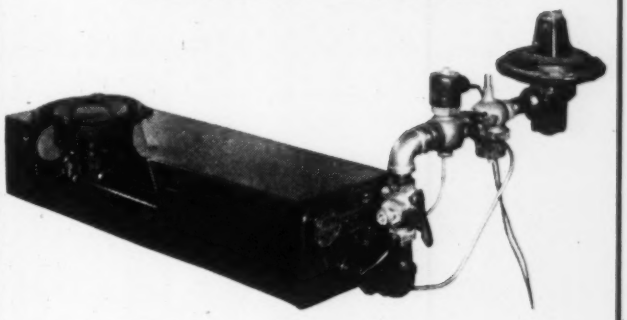

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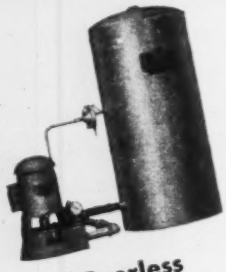

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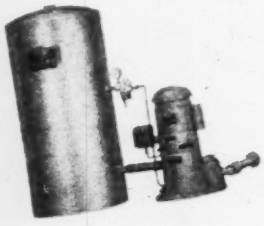

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# MEETINGS & CONVENTIONS

## ASHVE Meeting to Have Three Technical Sessions

In addition to various committee meetings and social activities, the presentation of 12 papers at three technical sessions has been scheduled by the American Society of Heating and Ventilating Engineers for its semi-annual meeting, to be held June 2-4 at the Hotel del Coronado, Coronado, Calif. (across the Bay from San Diego).

A. V. Hutchinson, 51 Madison Ave., New York 10, is society secretary and detailed information concerning the program and other features of the meeting is contained in the ASHVE Journal Section of this issue.

## ASRE Completes Program for 34th Spring Meeting

The 34th spring meeting of the American Society of Refrigerating Engineers, 40 W. 40th St., New York City, will be held in Los Angeles, Calif., June 9-11. This is the first national meeting of the society to be held on the west coast in 30 years.

Among the many technical papers scheduled for presentation are: Mechanism of Heat Transfer, Panel Cooling and Heat Storage, by C. S. Leopold; Surface Temperatures and Heat Flow for a Concrete Slab with an Imbedded Grid of Pipes, by Carl F. Kayan, Columbia University; Use of Refrigeration and Air Conditioning in the Growing of Plants, by F. W. Went, University of California; Biological Considerations in Hydrocooling and Precooling, by W. T. Pentzer, U. S. Dept. of Agriculture; Engineering Considerations in Hydrocooling and Precooling, by James C. Rear, Union Ice Co.; Water Treatment and the Control of Scaling in Evaporative Condensers, by Ralph M. Westcott; Air Flow Measurement in the Laboratory, by Daniel D. Wile, Carrier Corp.; and New Developments in Refrigerant Lubricants, by Edgar S. Ross, Sun Oil Co.

All ASHVE members are cordially invited to attend all technical sessions and to participate in the social activities.

## HPACCNA Meeting to Include Sessions on Panel Heating, Pipe Welding

The 58th annual convention of the Heating, Piping and Air Conditioning Contractors National Association will get under way at Cincinnati, Ohio, when the opening session is called to order at 2:00 p. m. Wednesday, June 11. The meeting headquarters will be the Netherland Plaza hotel, and the convention will continue through June 14.

In addition to the reports of various committees on standards, apprenticeship, boiler output, and air conditioning, as well as discussions of various problems of the industry, a number of technical sessions will be held. A forum discussion on radiant panel heating is scheduled and plenty of time for questions from the floor has been promised. Welding stainless

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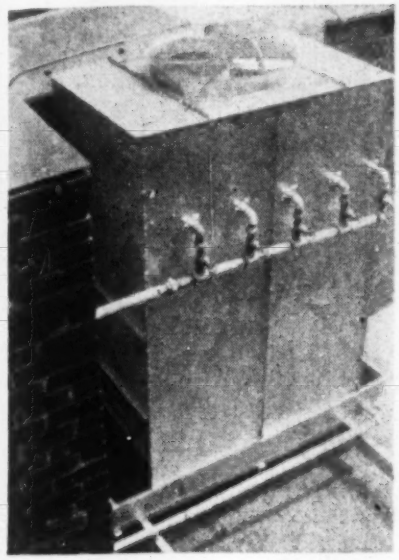

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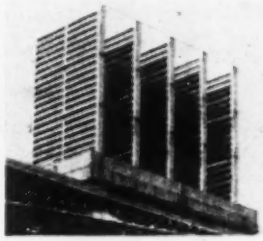

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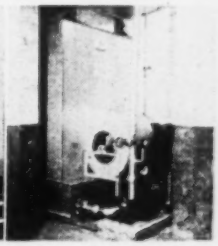

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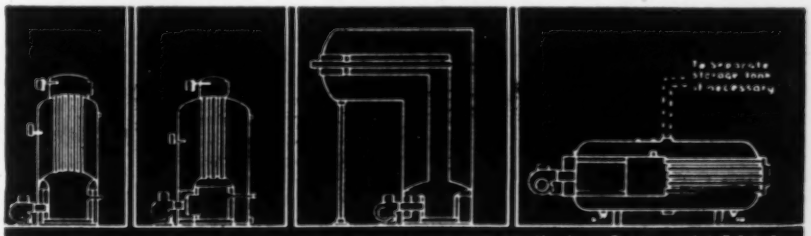

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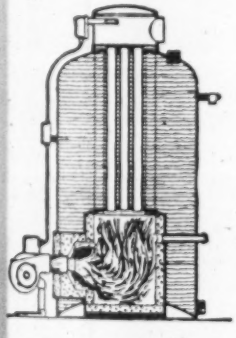

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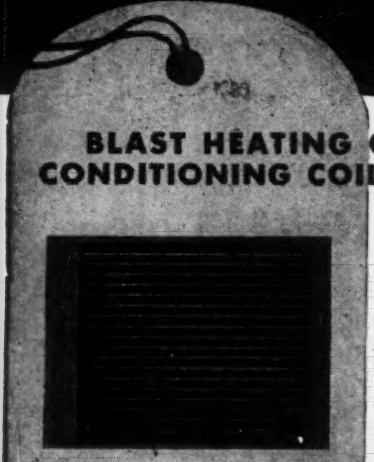

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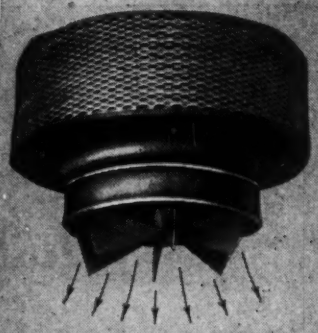

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steel will be a feature of the pipe welding session scheduled for Friday morning.

Association secretary is J. C. Fitts, 1250 Avenue of the Americas, New York City.

### Many Technical Papers, Inspection Trips Planned for ASME June Meeting

Opening at the Chicago headquarters hotel, the Stevens, on Sunday, June 15, the semi-annual meeting of the American Society of Mechanical Engineers, 29 W. 39th St., New York City, will continue through Thursday, June 19 with some 40 technical sessions, inspection trips, and entertainment features.

Technical sessions will start Monday morning. Six sessions will be devoted to power, four to management problems, three each to aviation, fuels, metals engineering, and oil and gas power. One or two sessions will be given over to heat transfer, industrial instruments and regulators, machine design, process industries, production engineering, engineering education, and citizenship.

### Plans for Stoker Manufacturers Meeting Nearing Completion

A recent announcement from the Stoker Manufacturers' Association, 307 N. Michigan Ave., Chicago, indicates that plans are well under way for the 1947 annual meeting which will be held at Lookout Mountain, Tenn. (near Chattanooga), June 16 through 18.

### NDHA Members to Meet in Atlantic City

The 38th annual meeting of the National District Heating Association will be held at the Chalfonte and Haddon Hall hotels, Atlantic City, N. J., June 24-27.

J. F. Collins, Jr., secretary of the association, has recommended that members request a room at the Chalfonte rather than at Haddon Hall as all functions except for the annual banquet, will be held in the former hotel. It is also suggested that members plan to arrive on Monday evening, June 23.

Association headquarters is at 827 N. Euclid Ave., Pittsburgh 6, Pa.

### OHI Plans Meetings, Elects New Officers

At the recent 24th annual meeting and convention of the Oil-Heat Institute of America the retiring board approved contracted arrangements for a convention and exhibition of products at the Coliseum in Chicago, the week of April 5, 1948, and unanimously authorized the managing director to proceed with preliminary arrangements for a convention and exhibition to be held on the east coast in the spring of 1949. The directors elected R. S. Bohn, president Preferred Utilities Mfg. Corp., New York, president of the institute. Vice presidents elected are A. T. Atwill, president, Quaker Mfg. Corp., Chicago, and C. R. Collins, vice president, Gerotor May Corp., Baltimore, Md.

Institute headquarters is at 30 Rockefeller Plaza, New York City.

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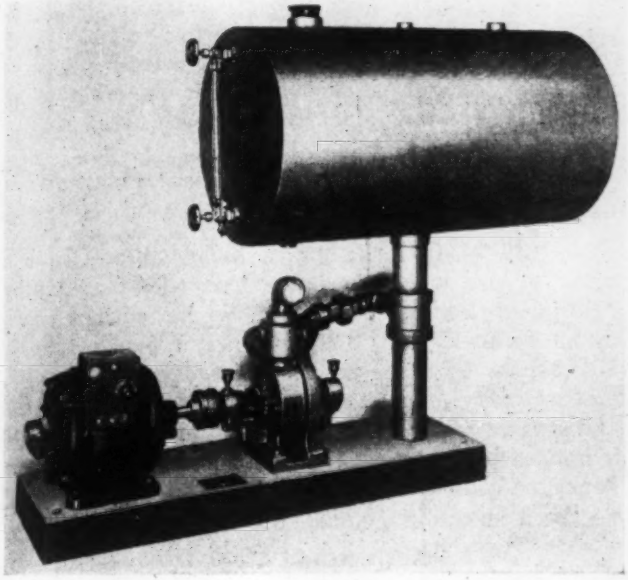

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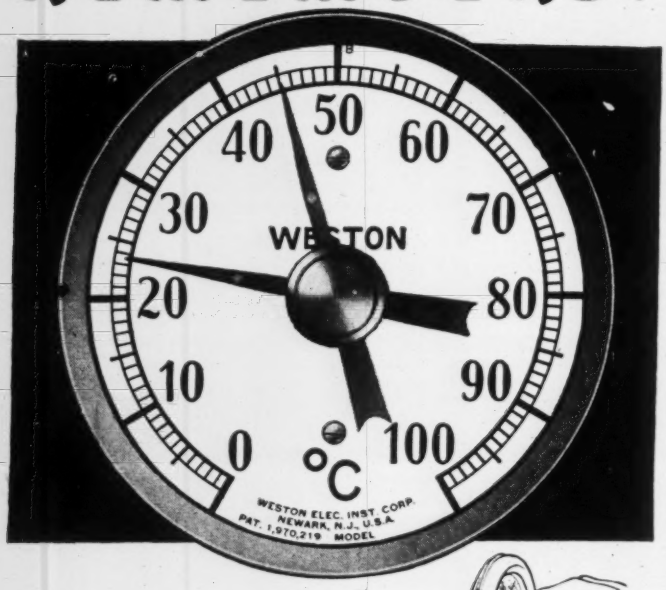


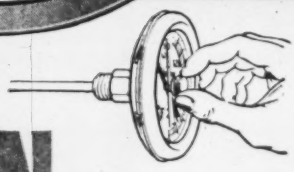

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The red index pointer on the MAX-MIN is manually set to either the low or high side of the main indicating pointer. When set on the high side, for example, it moves as the temperature increases, always remaining at the extreme temperature reached, until manually re-set. Thus in one quick reading you get both *present* temperature and the *peak* temperature since last reading.

For complete details see your jobber or local Weston representative. Weston Electrical Instrument Corporation, 648 Frelinghuysen Avenue, Newark 5, New Jersey.

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# RECENT TRADE LITERATURE

*For your convenience in obtaining copies of these bulletins, see coupon on page 171. If you write direct to the manufacturer, describe carefully what literature you want, as the number given first in each item is for use only when sending requests to Heating, Piping & Air Conditioning.*

### Air Compressors

No. 6497—Bulletin H-620-B26 released by Worthington Pump and Machinery Corp., Harrison, N. J., describes "Air King" compressors from 1 to 15 hp. The folder covers single stage units designed for pressures up to 100 and 150 psi. Two stage air compressors are offered for 125 to 250 psi working pressures.

### Air Conditioning and Refrigeration Controls

No. 6498—Air conditioning and refrigeration controls for use with ammonia, "Freon-12" and methyl chloride are covered in detail in the new catalog issued by the Hubbell Corp., 319 N. Albany Ave., Chicago 12, Ill. Among the items described are back pressure regulating valves, combination back pressure regulating and stop valves, dual pressure regulating valves, automatic suction stop valves, relief valves, strainers, gages, and solenoid valves for all refrigerants and for use with air, water, steam, brine, and oil.

### ASME Publications

No. 6499—The 1947 publication catalog has been issued by the American Society of Mechanical Engineers, 29 W. 39th St., New York 18, N. Y. Listing and giving prices of the technical material which may be obtained from the Society, the catalog covers pamphlets, periodicals, reports, technical papers, and books on numerous subjects including piping, hydraulics, and heat transmission. Boiler construction codes, American standards adopted and published by the ASME, and safety and power test codes are also listed.

### Boiler Water Conditioning

No. 6500—Boiler Water Conditioning at West Point Power Plant by F. R. Meyers, chemical engineer, American K.A.T. Corp., 331 Madison Ave., New York City, is the subject of a paper published in Power Plant Engineering and more recently reprinted by the author's company. Colloidal treatment, without inorganic salts, to remove old scale, reduce blowdown requirements and foaming tendencies is described in this case history.

### Centrifugal Pumps

No. 6501—Various types of centrifugal pumps, including single and multi-stage, direct and belt drive, and split case models are described and illustrated in the catalog of The Weinman Pump Mfg. Co., 290 W. Spruce St., Columbus 8, Ohio. Among the special units are sump pumps, nonclogging pumps, and condensate

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The handsome compact appearance of the TYPHOON self-contained air conditioning unit belies its rugged construction — an engineering achievement backed by over 30 years of practical experience.

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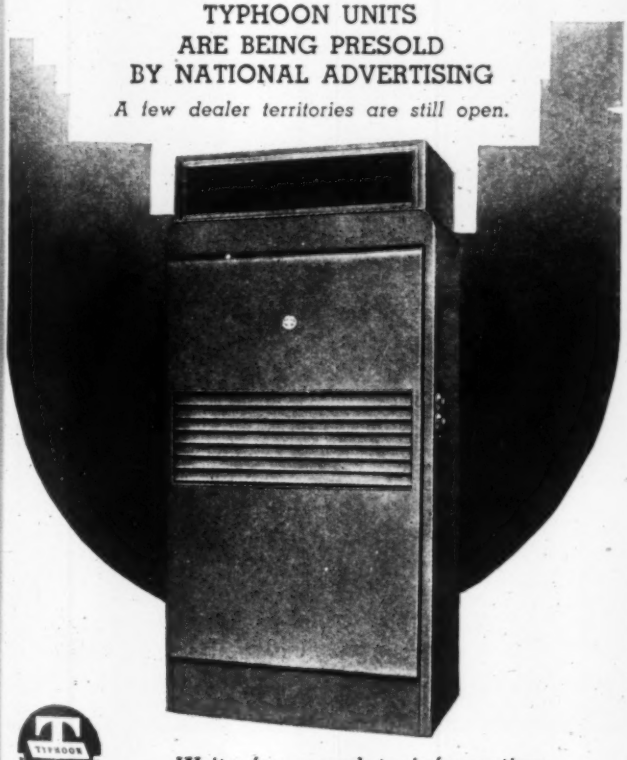

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*Division of Ice Air Conditioning Co., Inc.*

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Swartwout's New Powered Roof Ventilator

*The*

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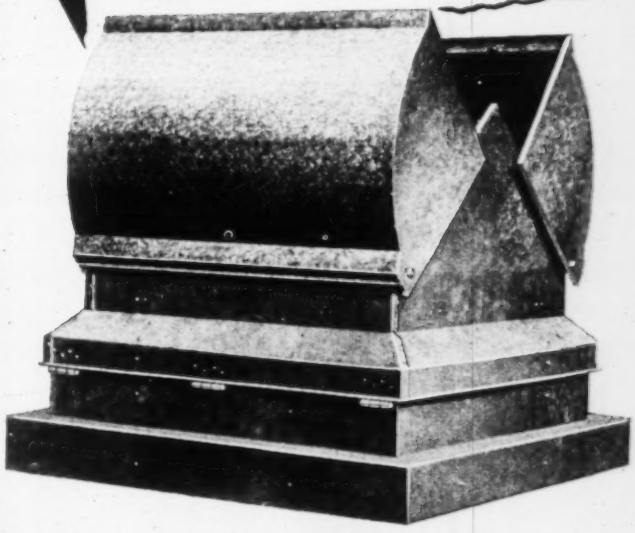

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For your smoke, fume, dust or heat problems that only *power* elimination will solve . . . this high velocity ventilator has many values for your consideration. . . .

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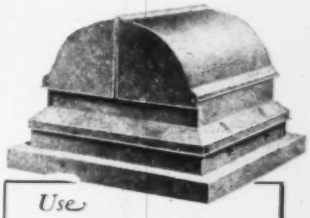
Carefully engineered in five sizes . . . and powered by motors and propeller type fans of varying ratings to give you a selection of 14 different capacities. Write for Ject-O-Valve Bulletin 323.



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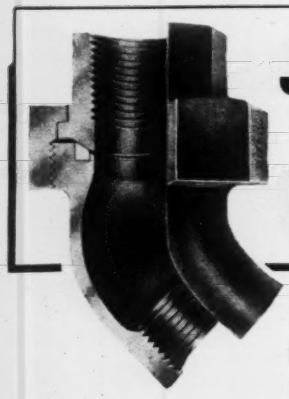

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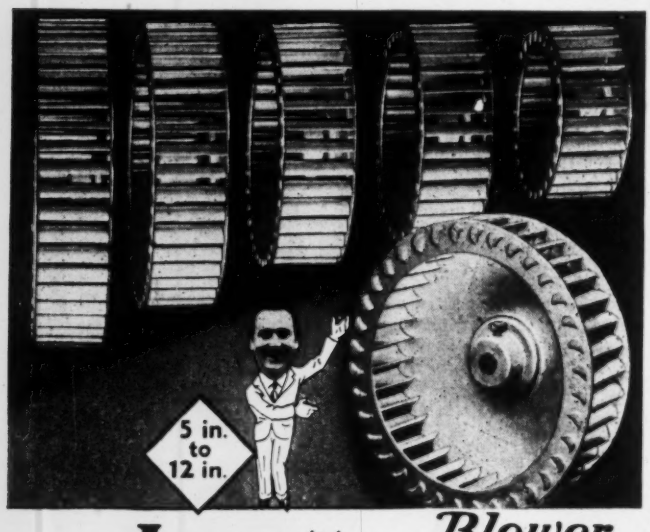


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**Janette steel wheels have been on the market for 18 years** and manufacturers of oil burners, coal stokers, blowers, generators, superchargers, dust collectors, hair dryers, gasoline engines, heating and ventilating equipment have used them with excellent results, which are possible because of their ***patented construction.*** The steel blades are cut in pairs, pressed thru slots in the heavy back plate, then welded. The tips of the blades are pressed thru slots in the inlet disc, then bent over against the inherent spring of the blades. This prevents loose blades and results in an exceptionally rigid wheel.

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return pumps. The more recent literature which has been added to the catalog covers Types GB and KB general purpose centrifugal pumps. The instruction manual contains a considerable amount of hydraulic engineering data.

### Condensate and Boiler Feed Pumps

No. 6502—Condensate return units and boiler feed pumps are illustrated and explained in a 24 page bulletin, No. CR-1, now available from the Deming Co., 14 Aetna St., Salem, Ohio. Detailed information on each type of unit is given and master selection tables simplify pump sizing.

### Continuous Specific Gravity Testers

No. 6503—Catalog section 91-B, released by Fischer & Porter Co., Dept. 1J-B, Hatboro, Pa., includes dimensions and operating details on an improved continuous liquid specific gravity indicator. Designed for continuous indication in closed piping systems, the unit is said to be used extensively in distillation work and in connection with all process systems where draining of sample batches is considered tedious or hazardous.

### Copper Tube Handbook

No. 6504—Completely revised with new chapters, the third edition of the Copper Tube Handbook for Plumbing and Heating has been published by the Copper and Brass Research Association, 420 Lexington Ave., New York City. The author, T. N. Thomson, is well known in the industry, having written a number of texts on plumbing, heating, and ventilating. The book covers the physical properties and types of copper tube, fittings, and the manipulation of tube and fittings. Also included are brief discussions of large building plumbing installations, the design of steam heating systems, and the design of hot water systems, with a few remarks on radiant heating.

### Deaerators

No. 6505—Four types of deaerators for four different services are described and illustrated in bulletin No. 3005 issued by the Cochrane Corp., Philadelphia 32, Pa. The deaerators covered include tray type units, widely used in industrial and central station power plants, atomizing units for handling particularly corrosive and incrusting water and for all marine service, cold water units for the elimination of oxygen without heating the water, and hot water generators and deaerators for protection of piping in hot water heating and service systems.

### Drilling Machine Spacer

No. 6506—The "Man-Au-Trol" spacer, a unit applied to radial or upright drilling machines for accurate spacing of holes to be drilled, reamed, and tapped, is designed to eliminate the need for special jigs. The semi-automatic unit is described in a 12 page bulletin released by the Bullard Co., Bridgeport

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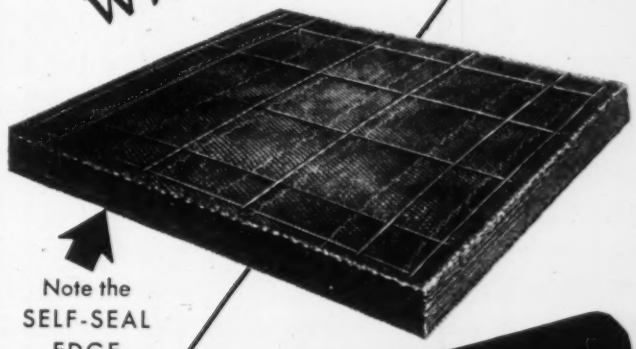

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## ARE EASIER TO SERVICE—CLEAN THE AIR MORE EFFICIENTLY

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2, Conn. A new booklet, Proved Interchangeability Without Jigs, also covers the advantages of this unit, and the company is offering to authorized groups the use of a 16 mm sound film on the spacer and another on turret lathes.

### External Pilot-Operated Pressure Controllers

No. 6507—Bulletin No. 462, a 20 page booklet, provides engineering, operating, and maintenance data on external pilot-operated pressure controllers manufactured by the Leslie Co., 51 Delafield Ave., Lyndhurst, N. J. Design features of the control pilots and associated diaphragm regulating valves are illustrated. Among the features stressed are the wide selection of operating media (air, clean gas, water, and light oils), noncontinuous leakage of operating medium, hard surfaced seat rings and 800 Brinell hardened stainless steel main valves, single seated, balanced construction, and the new "Conden-Seal" cooling bonnet. Selection charts and capacity tables are included.

### Ferrous Metallurgy Chart

No. 6508—Tempil Corp., 132 W. 22nd St., New York 11, N. Y., has published a 19 color chart, Basic Guide to Ferrous Metallurgy, which shows the working characteristics of steels at temperatures from –300 F to 2900 F. Hot working, annealing, normalizing, stress relieving, carburizing, and preheating for welding zones are shown. At the right of the chart 23 fundamental metallurgical terms are defined and ex-


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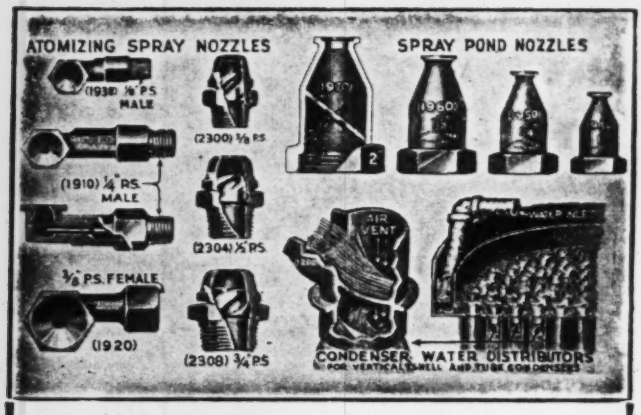


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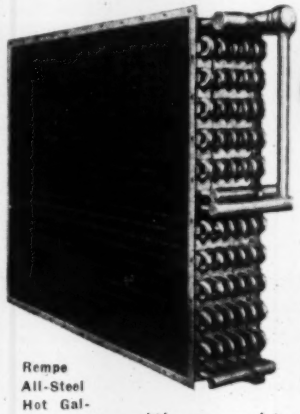

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When you need help in design and layout on coils for refrigeration and air conditioning take advantage of Rempe Engineering Service.

We can assist you by (1). Giving recommendations based on 58 years' experience in practical refrigeration engineering. (2). Design and manufacture of Fin or Pipe Coils to fit available space—to do a specific job.

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plained. The chart is available to industrial executives who request it on company stationery, stating their position.

### Flash Welding Pipe Line for Underwater Use

No. 6509—The flash welding of 3½ in. OD mild steel pipe in 4000 ft lengths and for use in a pipeline laid under the English Channel is briefly described in a reprint article featured in the Vol. 13, No. 4 issue of Flashes, the house organ of the Thomson Electric Welder Co., Lynn, Mass. The company's spot and roll seam welders are also described and illustrated in the issue.

### Heating, Ventilating, Air Conditioning, and Refrigeration Testing

No. 6510—Facilities for testing and rating equipment in connection with heating, ventilating, air conditioning, and refrigeration are described and illustrated in two new folders published by the United States Testing Co., Inc., 1415 Park Ave., Hoboken, N. J. Small steam boilers, room air conditioners, radiators, convectors, and refrigeration compressors and condensing units are among the items which the company is equipped to test.

### Industrial Controllers and Recording Instruments

No. 6511—A new bulletin (No. W-1811) listing over


**STEP UP your circulation with AUTO-VENT Air Eliminators**

Simple as A-B-C . . . yet positive in action everytime! Sure-cure for air pockets and air traps that hold up the free circulation of water or other liquids flowing through pipes—pressures to 75 lbs. Simple in construction—easy to install!

Built of brass with self-closing float operated valve. Withstands high temperatures, oil and other adverse conditions. Try these No. 7 Auto-Vents once, and you'll use them on every job. They're that good, yet NOT expensive!

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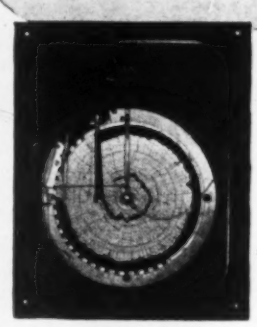

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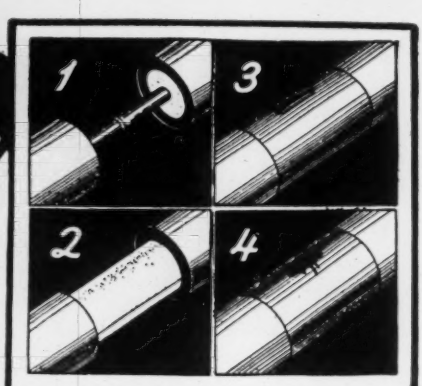


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For all your underground and exposed piping jobs you'll like DURANT Pre-Sealed Insulated Pipe. It is waterproof . . . reduces installation costs to a minimum . . . eliminates electrolysis and corrosion . . . requires very little trenching and field work . . . avoids extra cost of waste and breakage . . . comes to you ready to install. In short, D.I.P. is the *perfect pipe* for this type of job. Investgate D.I.P.! Write for details today.

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150 items maintained in stock for immediate deliver has been published by the Bristol Co., Waterbury 9 Conn. Recording thermometers, pyrometers, pressu gages, voltmeters, ammeters, and pH recorders ar listed. Other recording and indicating controller miscellaneous instruments, and accessories are al included.

### Industrial Thermostats

No. 6512—An extensive line of "Thermoswitch" thermostats for many commercial and industrial a plications is covered in a new catalog issued by Fe wal, Inc., Ashland, Mass. Principles of operatio price lists, and design, application, and construct features are included. A considerable amount of spa is devoted to information on lag and overshoot, proce characteristics, differentials and temperature vari tion, and checking control systems under operati conditions.

### Linear Conversion Table

No. 6513—A revised version of its linear tab which converts inches and fractions of inches decimal parts of a foot, has been issued by Babco & Wilcox Tube Co., Beaver Falls, Pa. The table (Tec nical Data Card No. 110A) is designed to save ti for those who deal with products such as tubing, ge erally sold and quoted on a per 100 ft basis.


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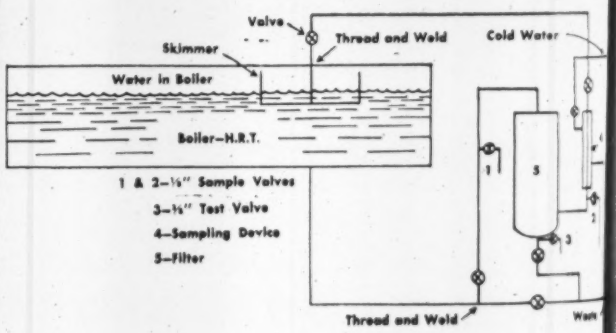
RODWELL'S BOILER WATER FILTER For HIGH or LOW PRESSURE BOILERS




Here's a simple and effective way to keep boiler water CLEAN. The Rodw boiler water filter helps to eliminate the troubles caused by scale, pi etc., which sooner or later result in an expensive retubing job or new boi

The "Skimmers" (shown above) draw the suspended solids from the t of the water, which passes through the Filter (#5) where all solids a foreign matter are removed and the water returned fresh and clean to t boiler. No clogging or scaling troubles . . . and no long drawn out t cleaning operations!

It will pay you to investigate this time and money saving device today. J drop us a line. Franchises are available for responsible dealers.

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### Nonadjustable Bimetal Thermostats

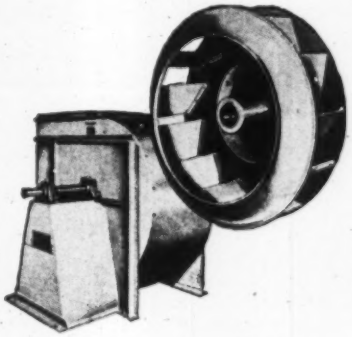
No. 6514—A four page folder describing a line of nonadjustable bimetal thermostats has been released by the Stevens Mfg. Co., Inc., Richland Trust Bldg., Mansfield, Ohio. Principles of operation, applications, dimensions, and calibration curves are included for these Type S thermostats which have a temperature range from 0 to 650 F and which are designed for 15 amp, 115 volt, a-c operation.

### Ozone Unit

No. 6515—The "Ozo-Ray" unit, designed to remove odors, smoke, and gases in the air, is described and illustrated in an eight page folder issued by Ozo-Ray Products Mfg. Co., Inc., 7 Dey St., New York 7, N. Y. According to the manufacturer, air is passed through an ultraviolet field created by the generating unit. The passing air is ionized and the electric field surrounding the elements further activates the air with the result that the oxygen molecules pick up an extra atom each. This ionized air, charged with added oxygen, is said to attack organic impurities. Units for single room installation as well as for incorporation in ventilation and air conditioning systems are described.

### Packaged Steam Generators

No. 6516—Recently published by Preferred Utilities Mfg. Corp., 1860 Broadway, New York 23, N. Y., is an

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**GUARANTEED EFFECTIVE** If you find SOL-VET "08" is not exactly as represented, when used according to instructions and full, accurate report is given within 30 days from date of purchase, your money will be *refunded in full*.

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**WRITE TODAY** for complete data and analysis and recommendations for your specific problems.

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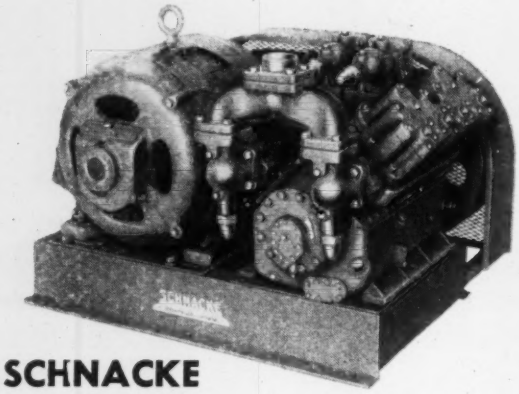

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All Schnacke units have replaceable refrigerant-cooled steel cylinder sleeves, complete pressure lubrication, balanced forged crankshafts, individual cylinder liquid relief, individual cylinder valve cage assemblies, built-in suction strainers, sealed crankcase, balanced pressure shaft seal, insert type bearings, water or air cooled heads, etc.

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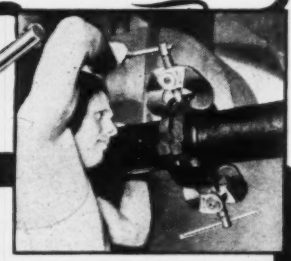

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**Flange-Jacks**


eight page bulletin (No. 1000-E) describing the company's unit steam generators. These self-contained oil-fired units are of four pass, induced draft construction, and range in size from 40 to 500 hp. They are designed for automatic operation on heavy oil with guaranteed thermal efficiencies of 80 per cent.

### Pressure and Temperature Regulating Devices

No. 6517—A number of new sections containing revised material have been added to the extensive catalog (Bulletin M) of the Ruggles-Klingemann Mfg. Co., 4 Foster Ct., P. O. Box 550, Salem, Mass. Electrically-operated valves and step action hydraulic regulators are but a few of the many industrial pressure and temperature regulating devices offered.

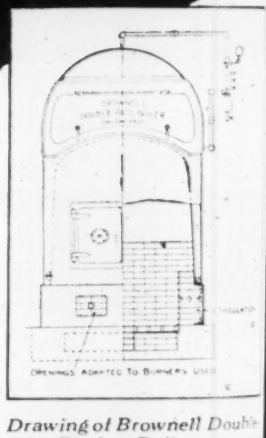
### Protective Coatings

No. 6518—The line of protective coatings offered by the Prufcoat Laboratories, Inc., 63 Main St., Cambridge 42, Mass., is described in a new four page bulletin. In addition to covering the chemicals which the coatings are designed to resist, the bulletin includes application data, and a variety of case histories. The coatings are applied with a brush to masonry, metal or wood and are designed to resist corrosion from acids, alkalies, alcohol, oil, and water.

### Pumps

No. 6519—A new catalog (No. G-47) of 180 pages


**BUILT FOR OIL OR GAS FIRING**



*Drawing of Brownell Double Pass Firebox Boiler arranged for oil or gas firing.*

Brownell Double-Pass Firebox Boilers are especially designed to burn volatile fuels. When so ordered, they are arranged for oil or gas burners.

Furnace volume and fire-tube systems in these boilers are balanced and generous to promote economical fuel consumption. Surpassing mere code requirements, Brownell engineers incorporate in the boilers many factors which increase efficiency and promote length of service.

Detailed information is given in our bulletin No. RM-1. Ask for a copy. When we answer, we'll give you the name of our nearest agent.

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has been announced by the Deming Co., 14 Aetna St., Salem, Ohio. The first 86 pages cover centrifugal, sump, drain, deep well, condensate return, boiler feed, rotary, and high pressure pumps. The balance of the catalog is devoted to the company's water systems for domestic and farm use and to pump accessories and spray pumps. Performance data, selection charts, and other pertinent information are included.

### Radiator Valve

No. 6520—The No. 44 "Lifetime" radiator valve offered by The Trane Co., La Crosse, Wis., is said to be the only valve of its kind with a lifetime guarantee against stem leakage. A simple bronze disc is used to seal the steam chamber from the outside and to transmit the motion of the wheel handle. It is described and illustrated in bulletin DS 260 and also in the Vol. 10, No. 7 issue of the company's house organ, Weather Magic.

### Refrigeration Load Data

No. 6521—Catalog No. R-114, published by Kramer-Trenton Co., Trenton 5, N. J., manufacturers of an extensive line of refrigeration and air conditioning equipment, contains basic data on refrigeration load calculations. While most of the data and examples given pertain to commercial refrigeration and cold storage applications, some of the information is of a general engineering type and is also useful in air conditioning design.


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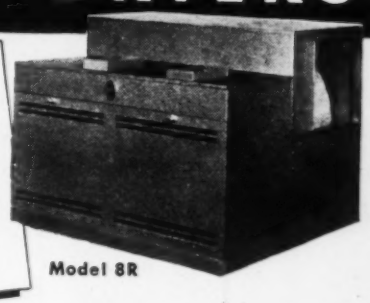

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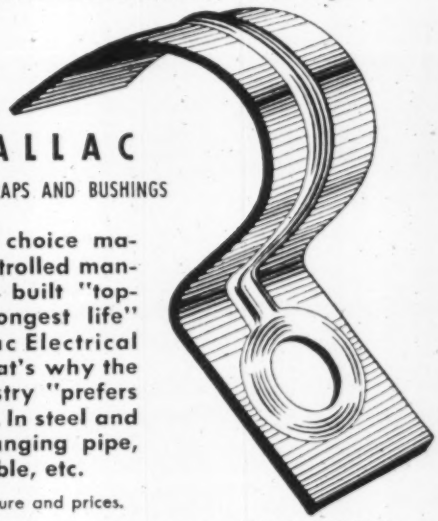

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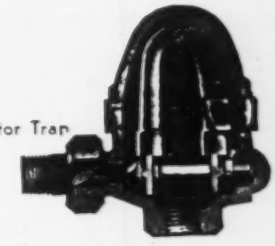

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